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THE THACHER PROCESS OF MOULDING PROPELLERS.

By Lieut. Comdr. D. F. Ducey, U. S. N., Member.

Previous to the developing of the subject process, which will be described in detail later, there were two general methods of manufacture in use. In the sweep method, a foundation plate is first cast and levelled in position, and the spindle set and secured. A layer of bricks on the plate is then covered with loam and the seating swept up. The center lines of the blades, to insure proper location of the blades with respect to the hub, are then laid off on the seating and the hub built up and shaped to form by means of a sweep attached to the spindle. The next step is the sweeping of the blades. The guide boards are placed and the piers built up of brick and loam mixed with a porous material; this being covered with loam, the driving face of the blade is swept up. The center line of the blade is then established from the seating and guide board, and the blade

shape marked off. Next the blade thicknesses at the several radii are established by wooden thickness pieces, these being secured to the driving surface. Parting sand is then placed over the swept-up surface and the space between the thickness pieces filled in with damp moulding sand, the top surface being made fair with the tops of the pieces to give a copy of the blade back.

The cope, or top half of the mould, consisting of a number of flat plates made to correspond with the back surface of the blade is clay washed to make the loam adhere to it, placed on the seating prepared and secured to the foundation plate, a thin layer of loam being laid over the image of the blade. The cope being completed and square marks made to insure its going back to its exact position, it is rolled and the sand pattern and thickness pieces removed. The surfaces being prepared, the mould is allowed to dry, after which the cope and drag are assembled, cores placed, etc., and the metal run.

The mould is then broken, after which the chipping of the blade backs, machining, polishing and balancing operations are undertaken.

In the pattern method the process is carried out in two ways. In the first, the pattern, of wood, for one individual blade with the hub or a portion of the hub is mounted on a spindle and the individual blade mould formed, the pattern will then be rotated on the spindle to the position for the next blade, etc. In the second, a solid pattern is made and the mould made up from this. The solid pattern is made generally of wood, but where quantity production is required, as in the case of our destroyers during the war, aluminum patterns were used.

The sweep method, being the oldest and most generally accepted, has been and is still used for wheels of all sizes. The use of the pattern method is more restricted being generally confined, in the service, to wheels below 12 feet in diameter.

Propeller blades are made of cast-iron, cast-steel, forged

steel, manganese or some other strong bronze, and monel metal. In solid propellers the hub is, of necessity, of the same material as the blades; but in the case of built-up wheels, while generally the same material is used for both blades and hub, for the poorer classes of work, when cast-iron or cast-steel blades are used, semi-steel is employed for the hub. In the service, manganese bronze having a tensile strength of 65,000 is specified.

In order that a propeller will function as designed, it is necessary that, in the finished wheel, the following conditions exist: The diameter must be true to pattern; the radius of any one blade must not vary by any appreciable amount from the radii of constituent blades; the pitch of each blade must be accurate and parallel with each corresponding blade, that is to say, any given point on one blade must be on the same plane with the relative point on each of the other blades; the blade areas must be identical and true to designed form, this entailing like widths at equal radii; there must exist a uniform distribution of homogenous metal both for the purpose of strength and of balance, this requiring like thicknesses of blades at corresponding radii; the blade and hub surfaces must be smooth.

From the above it is readily seen that the producing of a perfect propeller wheel is by no means an easy task. Assuming the wheel designed, the various processes in the old methods require skilled pattern work, moulding, metal mixing and running, assembling and extensive and accurate machining. Collectively, and if truly executed, these operations produce the finished product true to design. If, however, there be a failure to execute any single operation properly, or there be a slip in the co-ordinating of the several operations, and there are plenty of openings for such a slip, the finished wheel is imperfect. Whether or not the wheel becomes a total loss depends not wholly upon the magnitude, but rather upon the character of the error or errors, as the case may be. The

history of propeller casting, as in all other like processes, is the story of how man has tried to simplify the process and eliminate the chances for personal error. The more mechanical and simpler the method, the less chance for error and the nearer perfect the finished product.

However, rational as this last statement is, it appears that up until the late war, very few radical steps were taken along this line and propeller casting instead of being put on a scientific basis, with possible standardization in view, remained a series of operations for turning out the finished product, the grade of the finished product and the time for completing depending almost entirely on the individual dexterity of the personnel employed.

During the war necessity caused to be developed many new processes for the manufacturing of material necessary to its successful prosecution. Owing to the intensity and temporary success with which the belligerent submarine campaign was being prosecuted it became necessary to devote no small part of our effort to shipbuilding.

Previous to 1914, our shipbuilding being of no great magnitude, this extra demand for ship material, coming on top of that already created by the Allies, taxed the capacities of our manufacturers to such an extent that many old and established firms who had been doing kindred work, extended the scope of their activities to include the ship material most required.

Among these was the firm of Geo. H. Thacher and Company, now known as the Thacher Propeller and Foundry Corporation of Albany, New York. This firm, which for many years had been engaged in the manufacture of marine and other gray-iron castings, undertook to specialize in the manufacture of propellers and attacked the problem from a foundry point of view along rational but heretofore undeveloped lines, developing the process described herein following.

The Thacher method of casting consists in employing a separate and identical mould for each blade and locking the moulds together in accurately radial and properly spaced positions around their common center, the positioning of the blade moulds being obtained by means of a bed plate having a true machined surface and accurately positioned holes or bearings interlocking with similar surfaces and holes or projections carried by the separate moulds.

In order that the impressions of the blades in the moulds may be identical in shape and location the moulds are preferably, but not necessarily, formed in separate flasks from the same pattern or box, the flasks and the pattern box being all divided with each half of the pattern box designed to interlock with one-half of the flask. The flasks in turn interlock with the holes or projections of the base plate, so that if the base plate is accurately made and finished, the propeller will necessarily be symmetrical, provided the method has been properly followed. The flasks containing the blade moulds are set up on the bed plate around a bottom core attached to the center of the plate and fixed in position by means of bolts in the registering holes which determine their (the flasks) positioning. The meeting edges of the several moulds are then sealed with "slurry" to avoid fins on the finished casting and the mould closed with suitable center cores for forming the hub. The assembled mould is then ready for the metal.

As it is believed that a much clearer understanding of the manner of using the apparatus and of the processes involved will be had if the reader is familiar with the individual parts employed, these parts and their uses and functions will be described in detail.

ASSEMBLY PLATE.

The bed, base or assembly plate (see Figures 4 and 6) for small wheels is a sound iron casting, the top surface of which is machined to a true plane. For larger sized wheels, it is

made in two or three parts. This plate, as it must always remain true and not be subject to distortion, is either quite thick or, when it is desirable to reduce weight, it may be constructed of closely spaced bars, sufficiently deep to insure rigidity and having the upper face machined off to a true surface. The base plate is always of greater diameter than the wheel to be cast; however, this does not necessarily mean that for each size wheel a distinctive plate is necessary, as actually but two or three different sized plates are used, these taking all propellers from those required for small boats up to and including wheels twenty feet in diameter. While this latter figure represents the maximum size that may ordinarily be expected, there is nothing to prevent the use of plates suitable for wheels having a greater diameter.

The assembly plate having been machined and its trueness verified, a hole (see Figure 4) is drilled in the exact center; this hole is for the purpose of properly locating the jig used when drilling the holes for the flask assembly. These holes are, those in the assembly plate itself, the nowel pattern plate, cope-container, nowels and copes; these latter parts will be described later. Around this central hole is provided a circular recess (see Figure 4) having a depth equal to one-half the thickness of the assembly plate, for the purpose of receiving a boss formed at the center of the under side of the centering plate. By means of the jig mentioned above, at equidistant points around the circumference of the base plate, there are located a series of holes (see "*a-a*," Figure 6) extending out radially from the center. For a three-bladed wheel there must be three sets of these radial lines of holes, each set of radial lines of holes being exactly 120 degrees from each of the other sets; in the case of a four-bladed propeller, there must be four of these sets of radial lines of holes, each of these radial lines of holes being exactly 90 degrees from each of the other three sets.

The number of holes in each set is six and their distance apart depends on the size of the flask used. The location of the six holes is as follows: The center hole on the assembly plate is considered the hub locating hole for each set, the second hole is located at a radius greater than the radius of wheel cast in order to fix the outer edge of the flask (consequently wheel diameter and blade angle), and the remaining four holes are divided two for each side of the flask to assure perfect, immovable and constant alignment of each set. From this it will be seen that the sets of radial lines of holes in the assembly plate serve several purposes:

(*a*) They insure that each flask—that is, each blade—will be exactly located with respect to each other blade at the predetermined angle, 120 degrees in the case of a three-bladed wheel and 90 degrees in the case of a four-bladed wheel.

(*b*) They insure that each flask, providing the flasks are similar, which they are, will be located at exactly the same radial distance from the center of the assembly plate: that is, provided the blades in the flasks are alike, a series of points on the tip of any one blade will be at exactly the same distance from the center of the assembly plate as similar points on each of the other constituent blades.

(*c*) They insure that any number of like flasks and blades may be consecutively placed in the same position (it will be shown later that it is impossible to do otherwise).

So much for the assembly plate for the time being.

JIG OR TEMPLET.

Closely allied with the assembly plate is the templet. This, while the least imposing part of the outfit is in reality, in spite of its plainness and simple construction, the foundation stone, as one might say. A mistake in laying out this leads to dire consequences, but once properly laid out the remaining steps in the process are so simple as to be classed as mechanical. The templet, of cast-iron sufficiently heavy to withstand ordi-

nary handling, is built from the drawing and after being machined to a true surface has drilled in it, as near to drawing dimensions as is humanly possible, the locating or positioning holes that are to be transferred to the assembly plate, cope-container, nowel pattern plate, nowels and copes.

To illustrate how it is employed and the necessity for accuracy, its use in laying out the radial holes in the assembly plate will be described: The assembly plate, which it will be remembered is machined true, is leveled and the jig laid on it so that the pivot centering hole of the jig corresponds exactly with the centering hole of the assembly plate, a machined dowel being used for this purpose. The jig is now firmly secured to the assembly plate and each locating hole in it accurately transferred to the assembly plate. This being completed, the jig is unfastened and if the locating holes are being laid out on the assembly plate for a three-bladed wheel, the jig is revolved around the dowel pin exactly 120 degrees and the process repeated. The third set of locating holes is laid out in the same manner. Before drilling, in order to verify the angular distance between the holes and insure it being 120 degrees in each case, the whole is trammed and if there is no discrepancy, the locating holes are drilled.

It appears that it would not be amiss here, before going ahead with the description of the various other parts, to briefly set forth the transfer principle involved in the process. With the assembly plate drilled as just described, take three identical jigs, assume each a mould of a propeller blade with a section of the hub; to properly locate each of these on the assembly plate so that they together will form a completed propeller mould simply requires that each be placed on the assembly plate, so that a dowel can be placed through each of the holes in the jigs and the corresponding holes in the assembly plate. If the dowels don't go through both holes, we know the individual moulds are not in their proper positions; but once the dowels do go through the matter is settled. Moreover,

the individual jigs being alike, there is no reason why numbers one and two cannot be lifted and interchanged. Once the locating holes are laid out properly, the proper locating of the individual blade and hub section moulds of a complete propeller mould becomes a fool-proof operation.

THE PATTERNS.

Referring to Figure 1, "A" and "A-1" are two accurately machined plates drilled to jig, the same as is the assembly plate, so that the holes "H-H" exactly correspond when "A" and "A-1" are superimposed on each each other, and likewise holes "H-H" will exactly correspond with the locating holes in the assembly plate if plates "A" and "A-1" should be placed thereon, all having been drilled from the same jig. "B" and "B-1" are two of the three sections of the pattern, being known respectively as the nowel pattern block and the cope pattern block. These are pitch surfaces mechanically made of hardwood, care being taken that the material used is such as will prevent shrinkage due to laying around, or deformation due to ramming up. Each of these patterns is built up on a plate such as "A" and the pitch plane developed as shown at "B." In the case of "B-1," the pitch plane is first established, as in the case of "B," and then the thickness of the blade is built up, this forming the third section of the pattern. It will be thus seen that if before the blade is developed, and the hub portion of the patterns be disregarded, the plates "A" and "A-1" be brought into contact, "B" and "B-1" will form a solid block with the pitch surfaces in perfect contact. Please note this carefully. The thickness of the propeller blade, which is really the pattern proper being noted before as the third section of the patterns, is marked "D," this may be either an integral part of "B-1" or be made separately and firmly secured thereon.

NOTE: The left-hand picture of Figure 1 is rather misleading. The machined plate upon which "B-1" is built up

is not shown, being underneath and forming the bottom plate of what is known as the cope-container, the plate or rather the flange "A-1" being parallel to the bottom plate. For the cope-container, the function of which will be explained later, see Figure 7.

The location of the pattern or pitch blocks on the plates "A" and "A-1" is fixed by jig.

THE COPE-CONTAINER.

This is shown in Figure 7, and while the term cope-container is generally used, it is believed that cope-pattern-block-container better describes it. The bottom plate, which the cope pattern block is built upon as noted previously, having been drilled to jig to locate the cope pattern block, is also drilled for attaching the sides of the container, bolts being used for this purpose after the flanges of the sides have been machined true. After the top flanges of the sides are true to the bottom flanges—that is, true to the bottom plate—locating holes are drilled in the top flanges (see "H-H" from the jig) for positioning the copes. The trunnions at the sides are for rolling. The blade form is here shown very clearly. The volume of the container is equal to the volume of the nowel pattern block plus the volume of the cope pattern block, without the blade pattern.

NOTE: This container does not form any part of the assembled mould.

NOWELS OR DRAGS AND COPES.

Figure 2 shows a nowel mould, "A," and a cope mould, "A-1," as made from patterns "B" and "B-1," respectively (see Figure 1). These two flasks are machined accurately; "E" and "E-1" in the nowel being parallel machined surfaces drilled to the same templet as are the holes "H-H" in plate "A," Figure 1. Surfaces "E-2" and "E-3" of the cope are also parallel machined surfaces with holes "H-H" drilled to the same templet as are the holes "H-H" in plate "A-1" (see Figure 1).

The nowel and the cope shown in Figure 2 are for small blades. For larger blades the construction differs somewhat; see Figure 9, which shows three nowels; note the sand carrying bars assembled on the base plate, center arbor and two copes. As can be seen these nowels are open at the top to set over the pattern block, with cross-bars for sand carrying and strength. The flanges, in order to prevent possible deformation, are made exceedingly heavy.

CENTER ARBOR, ARBOR CAP AND CENTER PLATE.

To the left of the nowel, in Figure 8, is shown the center arbor, arbor cap and center plate, "*a*," "*b*" and "*c*," respectively. The center arbor is a hollow iron casting which carries the center core. At each end is a turned bearing, one centering in the arbor cap and the other in the center plate; the latter, it will be recalled, being centered in the assembly plate.

A cast-iron ring, "*b*," with the center attached by brackets at the top, forms the arbor cap. This is placed over the center arbor, "*a*," and on to dowels for positioning on the assembled mould. It is made up with sand so as to carry a head of metal on the hub riser (see Figure 10).

The center plate, "*c*," is a small machine finished cast-iron circular plate with a boss at the center of the under side for locating it on the assembly plate, and a countersunk hole for fixing the position of the center arbor. The small projections shown on the upper face facilitate attaching the core sand.

The center arbor, arbor cap and center plate just described are only used in the larger work; in smaller work, as illustrated in Figures 1 and 2, the apparatus used is shown in Figure 5. "F-1" represents a machined turned core carrying plate, complete with core, which is centered in the assembly plate. The core, "J," sets in the core, "F-1," and also sets into and carries core, "K," which is the top center core. The holes, "L-L-L," in core, "K," are used as risers.

POURING RING.

As the metal is poured to the blade tips, a pouring ring, which eliminates the use of sumps, as shown in Figure 10, is fitted. This ring is a circular cast-iron trough which feeds the gates to each blade. Figure 10 shows the assembled mould for an eleven-foot wheel.

Summing up, it will be seen that the apparatus, for a three-bladed propeller wheel, consists of—

(*a*) Three nowels and three copes.

(*b*) One nowel pattern block, mounted on a machined pattern plate, with its surface swept up to the desired pitch.

(*c*) One cope pattern block, constructed similarly to the nowel pattern block.

(*d*) One blade form constructed of wood and secured firmly to the face of the cope pattern block.

(*e*) One cope-container.

(*f*) One assembly plate.

(*g*) One center plate, core arbor and core arbor cap.

(*h*) One pouring ring.

(*i*) Necessary securing bolts, buts and centering dowels.

(*j*) Sand and other moulding materials.

(*k*) One jig.

The process is essentially a die-casting one, the procedure followed being practically the reverse of that ordinarily employed. It is applicable to any size wheels and is equally applicable to the casting of solid wheels or separate blades. Briefly, it consists of the establishment of the designed pitch surface and a series of mechanical transfers of this surface; with the apparatus previously described this is accomplished as follows:

Referring to Figure 1, the nowel pitch or pattern block, "B," is located centrally on the plate "A" by jig, and the pitch surface swept up, thus establishing the pitch plane. The cope pitch block, "B-1," located centrally by jig on a plate

which forms the base of the cope-container (see Figure 7), is built up on "B" so that the pitch plane of "B-1" is the same as that of "B." The cope-container, Figure 7, is then built up around "B-1" with the top flanges, "A-1," parallel to the plate on which "B-1" rests. The height of the cope-container is such that, with the hub sections removed from "B" and "B-1" and "B" superimposed on "B-1," the pitch surfaces "B" and "B-1" and the surfaces "A" and "A-1," holes "H-H-H" being established from the same jig will make perfect contact throughout. Note that this is before the blade form is attached to B-1. The nowel, drilled to the same template as "A," is now placed over "B" and secured to "A" and rammed up; a true plate is now secured to act as a follower plate, the nowel rolled and the pattern block, "B," still attached to "A," withdrawn. Thus the pitch plane, "B," has been transferred from "A" to the bottom plate of the nowel, and as all nowels, for any one propeller, are of exactly the same height, the centers of all constituent blades will lie in the same horizontal plane. The nowel is then located on the assembly plate by placing body fit bolts through the holes in its bottom flanges and the holes in the assembly plate, it being recalled that all these holes were drilled to the same templet. The pitch surface has now been transferred to the assembly plate, and what is more, all blades are in the same horizontal plane, at the same height above the base of hub, and at the predetermined angle. For assembled nowels see Figure 4.

To establish the back of the blade, the blade pattern, "D," of Figure 1 (only one is necessary), is firmly secured to "B-1" and the cope, which is drilled to the same templet as "A-1," is secured by body fit bolts to "A-1," the top flange of the cope-container. The sand is rammed up and the cope-container, with pitch block "B-1" and pattern "D," withdrawn. The cope is now placed on top of the nowel and as the holes in its flanges are drilled to the same templet as are the holes in the flanges of the nowel, when the body fit bolts are inserted and drawn up, it being remembered that the volume of "B"

plus " B-1" is exactly equal to the volume of the cope-container, the blade mould is completed (see Figure 3 for assembled cope and nowel).

The above description of the process having been for the purpose of setting forth the transfer steps involved, it is considered well to describe the process as it actually takes place in order that an idea of the simplicity of the foundry steps involved and the limited amount of time necessary may be obtained.

After the assembly plate, Figure 6, has been placed on the foundry plan, the nowel pattern plate " A," of Figure 1, with the nowel pattern block, is set on a substantial floor and the nowel " A," of Figure 2, placed over it and securely bolted thereto. A suitable, strong, dry, sand facing is then riddled on the pattern block. A weaker dry sand mixture is then rammed in until the nowel is flush. A machine-finished follower plate is now bolted to the nowel and the whole rolled. The plate " A," Figure 1, with pattern block " B," is now unbolted and withdrawn, guide pins being used. The nowel is then ready for baking.

On moulding the cope the same operation is conducted.

The cope and nowel are then baked for from 16 to 18 hours in an ordinary core oven at temperatures varying from 360 degrees to 420 degrees F. After baking they are ready for assembling.

It will be noted that the cope can be made up simultaneously with the nowel and entirely independent therefrom. Each can be made up miles apart with perfect assurance that they will fit together when assembled.

After baking the nowels, the assembly plate having been cleaned off, each nowel is placed on it and bolted in position. The baked copes are then secured to the nowels by means of bolts fitting through the positioning holes, which are drilled to jig. The center plate "*c*," of Figure 8, is lowered into place on the assembly plate, the center arbor "*a*" entered into

the center plate and the arbor cap "*b*," set on top. All the joints are then closed and the pouring ring placed.

The mould is poured through gates to each blade, the metal flowing through each blade mould to the hub, rising here through the hub riser until sufficient head to insure solidity is provided.

To make up three nowels and three copes for a destroyer wheel takes about six hours moulding time; the assembling on the assembly plate requiring around three hours.

When the method of making the wheel is considered, it will be seen that after taking it from the mould, removing the fins on the blades and brushing up (grinding may be undertaken if a polish is required), the following conditions, as has been demonstrated in actual experience, will exist:

The pitch of each and every blade will be exactly true to the pitch plane established, as the transfer throughout has been mechanical; all blades are made from the same pattern and, owing to the depth and solidness of the pitch blocks no deformation of pitch face is possible.

The pitch face being exactly reproduced no machining of it is necessary, thus preserving the dense skin of the casting and avoiding the local strains set up by the use of tools. The back of the blade being moulded to form requires no work on it.

Machining of the hub for fitting purposes is necessary—that is all the machining involved.

Provided there are no blow-holes, which there should not be due to the method of pouring and venting, the wheel will be in perfect enough balance to warrant, without any further work on it, its use for the highest speed work met with today; as the blade angles are mechanically set, each blade is an exact reproduction of each other blade, and the radial distance of like points on constituent blades is established mechanically. Blade locations are as accurate as it is possible to transfer them from a jig.

The wheel diameter will be true to design, within the small limits of inaccuracy that comes from casting thin-edged blades, due to the establishing of the positions of the blade tips by jig.

Any given point on one blade will be in the same horizontal plane as the relative point on each of the other constituent blades, that is, no one blade leads or lags behind any of the others, because the same pitch plane established at a fixed height, is used for all constituent blades and the nowels are of the same height.

The transfer of the blade form being mechanical, the blade areas will be exactly similar and true to design except for small differences due to slight irregularities that may occur along the fine edges.

This process, in addition to being capable of producing a propeller as nearly true to design as is considered possible, has other merits that are of the greatest import.

First: It permits of standardization. For wheels built integral and for commercial use the equipment is built in three standard series, 12, 16 and 20 feet in diameter. Each unit is designed for either three or four-bladed wheels. The same standards are maintained for built-up wheels.

The patterns are standardized in diameter to suit the equipment diameter capacity, and in pitch to standard practice. That is, the patterns for the 16-foot equipment would be of sufficient length to fit the container and hence could be used on any diameter from 12 to 16 feet. For ordinary commercial work the pitches on the pattern blocks can be standardized for 6-inch pitch variation without introducing excessive modification of the blade form. In the case of high speed work, as in destroyers, this pitch variation could be reduced to three inches, in which case the necessary modification of blade form would not be prohibitive. Our present knowledge of propeller design makes such standardization possible.

Second: It permits of quantity production at rates and costs that a few years ago were considered unbelievable.

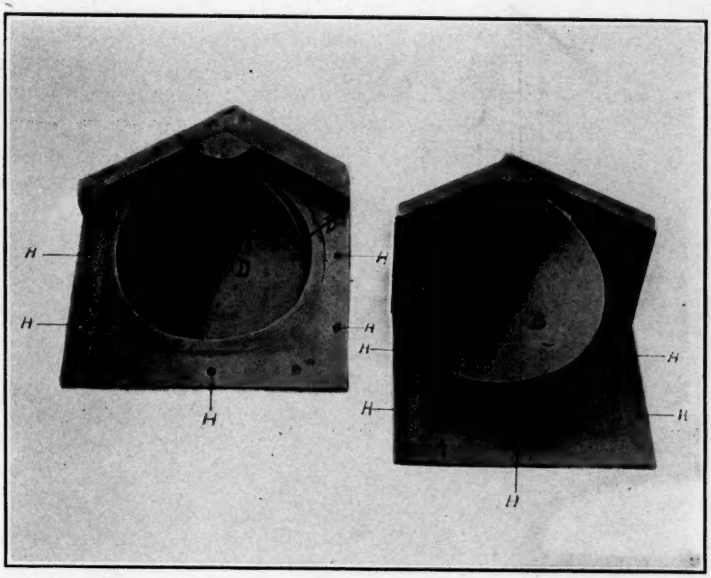


FIGURE 1.

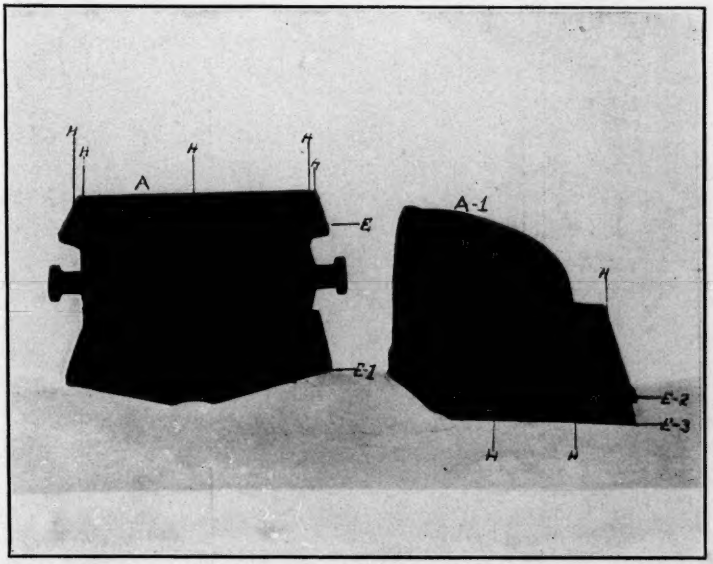


FIGURE 2.

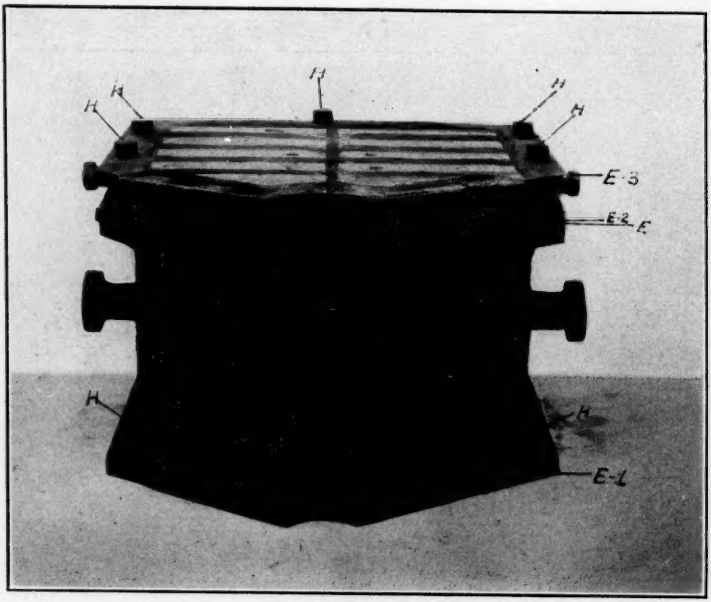


FIGURE 3.

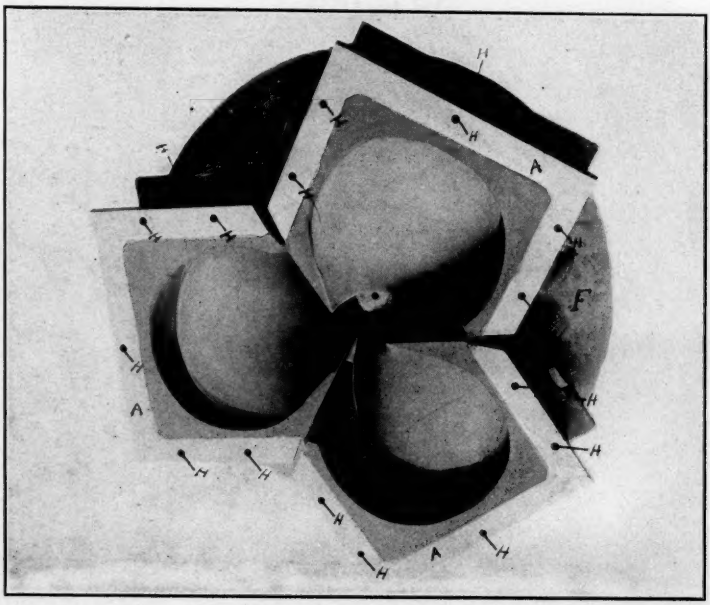


FIGURE 4.

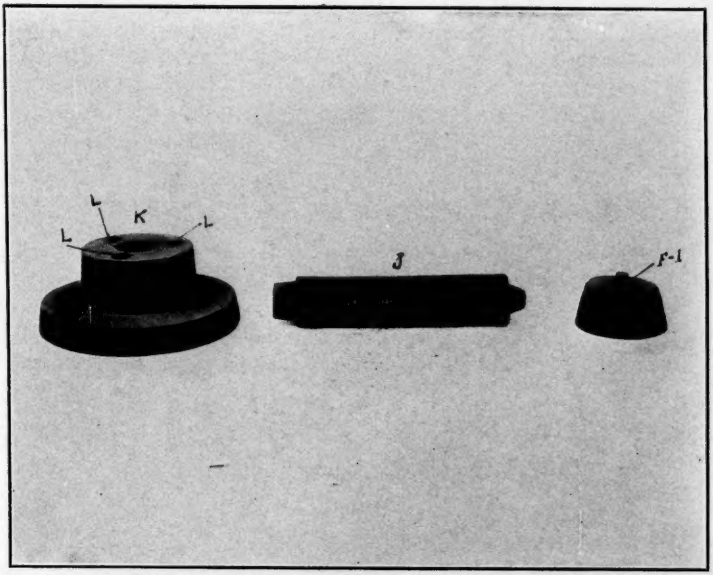


FIGURE 5.

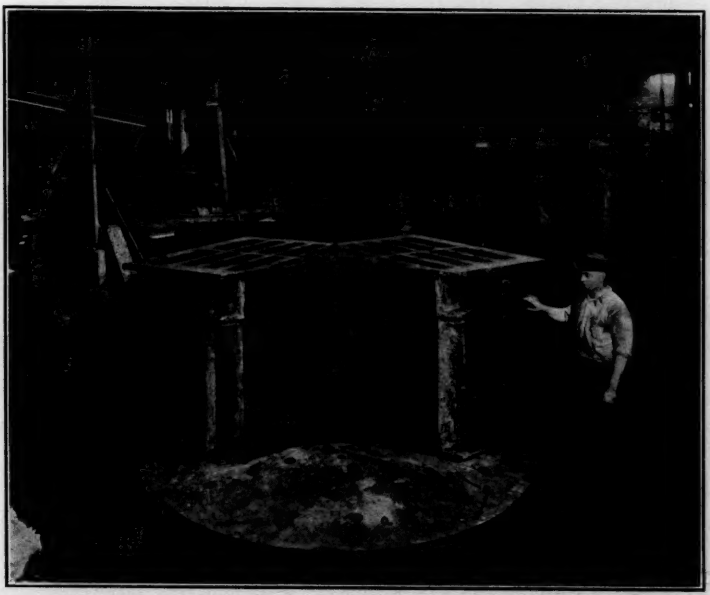


FIGURE 6.

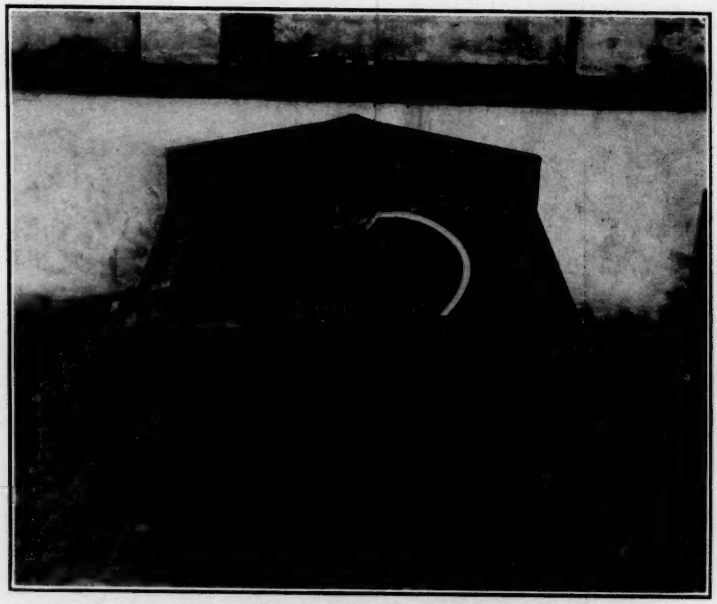


FIGURE 7.

FIGURE 8.

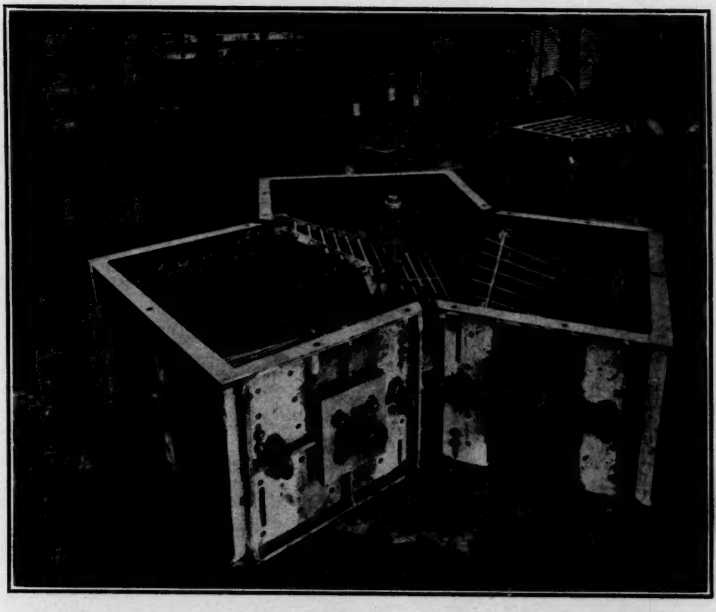


FIGURE 9.

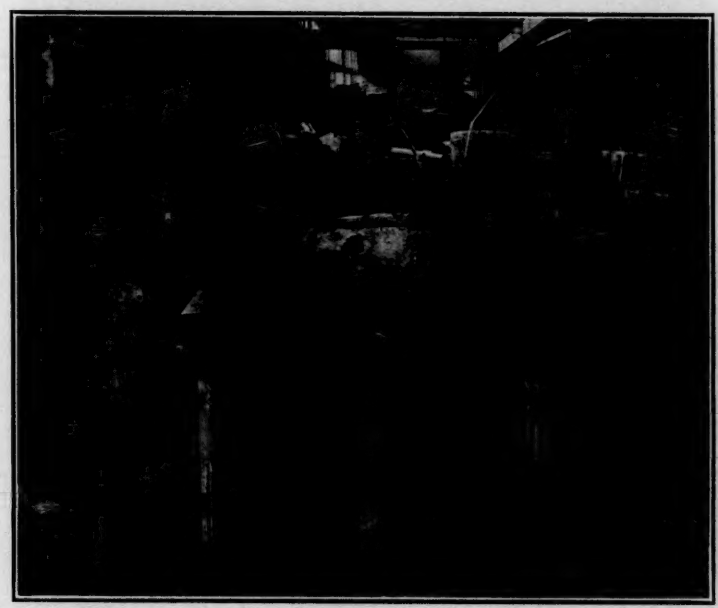


FIGURE 10.

Third: It permits of exact reproduction.

Fourth: And highly important, especially in these times, it does not require the services of skilled moulders. The moulding being purely mechanical, any ordinary intelligent man can be trained in a few days.

Fifth: While not yet developed there is no reason that the process cannot be used in burning on metal to broken blades. With repairs made this way there would be no doubt as to the accuracy of the repaired blades.

Sixth: While in the pattern method at present in general use it is necessary to charge from 50 per cent to 80 per cent of metal in excess of the finished weight of the wheel, the amount increasing as the efficiency of machining decreases, to allow for machining, the excess of metal required in this method is only 25 per cent to 35 per cent, and once the amount of excess is determined, no change, due to machining conditions need be made, as machining of the blades is unnecessary and can, therefore, be disregarded in the charge calculations.

Seventh: In addition to saving metal as noted above, the fact that no machining of the blades is required, eliminates the employment of machines, tools and machinists and results in decreasing the time of manufacture, in the case of a large wheel, by two or three weeks.

This process, the Thacher, based on sound mechanical principles in permitting the making of propeller wheels true to design, will be an invaluable aid in the furtherance of propeller design.

FIREROOM CONTROL ON THE U. S. S. *TENNESSEE.*

By J. H. King, Associate.

The present method of segregating each boiler on capital ships in a separate water-tight compartment presents a problem of operation that is quite different from that heretofore encountered. The older ships fitted with oil fired boilers have three firerooms, with three to four boilers in each. In these ships it is comparatively easy to get from one fireroom to the other and boiler room control is therefore a fairly simple matter.

The battleship *Tennessee* is the first ship in our Navy to be completed in which each boiler is located in a separate fireroom. The new battleships *Maryland* and *California* are similarly arranged, as well as all capital ships now laid down. Boiler operation on these ships is more complicated and necessitates handling the boilers from a central control station. Under these circumstances extreme flexibility and ease of control is essential. The method of fireroom control worked out for the *Tennessee* proved very satisfactory and it is believed that the experience gained on that ship will be of interest and value.

EQUIPMENT.

The boiler rooms are located outboard of the machinery space. Four firerooms are located on the port side and four on the starboard side as shown in Figure I. As stated above, each boiler is located in a separate water-tight compartment. Access from one fireroom to another is gained only by passing up a ladder from the fireroom floor, through an air-lock, to the second deck.

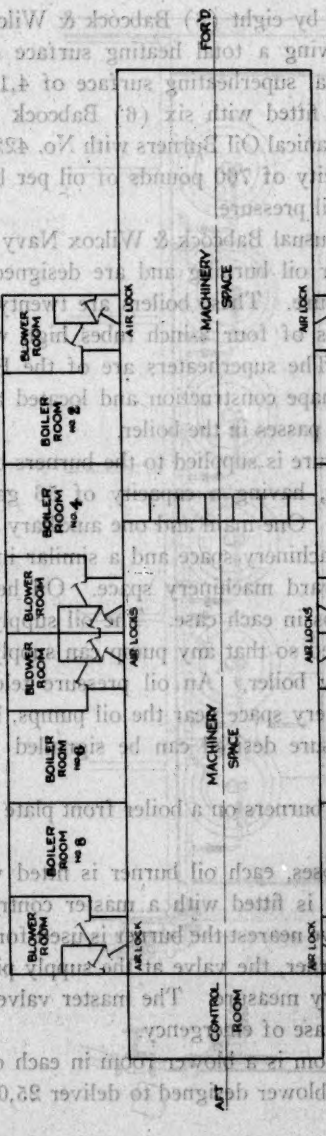

FORD
MACHINERY SPACE
AIR LOCK
AIR LOCK
BLOWER ROOM
BLOWER ROOM
BOILER ROOM NO 2
BOILER ROOM NO 1
BOILER ROOM NO 4
BOILER ROOM NO 3
BLOWER ROOM
BLOWER ROOM
AIR LOCKS
AIR LOCKS
BLOWER ROOM
BLOWER ROOM
BOILER ROOM NO 6
MACHINERY SPACE
BOILER ROOM NO 5
BOILER ROOM NO 8
BOILER ROOM NO 7
BLOWER ROOM
AIR LOCK
AIR LOCK
BLOWER ROOM
CONTROL ROOM
AFT

FIG. 1.

Steam is furnished by eight (8) Babcock & Wilcox boilers and superheaters, having a total heating surface of 41,768 square feet and a total superheating surface of 4,168 square feet. Each boiler is fitted with six (6) Babcock & Wilcox *Cuyama* Design Mechanical Oil Burners with No. 4220 sprayer plates, having a capacity of 760 pounds of oil per burner per hour at 300 pounds oil pressure.

The boilers are the usual Babcock & Wilcox Navy type with furnaces arranged for oil burning and are designed for 285 pounds working pressure. These boilers are twenty-nine sections wide, ten groups of four 2-inch tubes high with tubes 8 feet 0 inch long. The superheaters are of the Babcock & Wilcox standard U-shape construction and located at the top of the first and second passes in the boiler.

Fuel oil under pressure is supplied to the burners by turbine driven Kinney pumps, having a capacity of 56 gallons per minute at 2,400 r.p.m. One main and one auxiliary pump are located in the after machinery space and a similar installation is located in the forward machinery space. Oil heaters are placed above the pumps in each case. The oil supply lines to the boilers are arranged so that any pump can supply fuel oil to the burners on any boiler. An oil pressure telegraph is located in each machinery space near the oil pumps, by means of which the oil pressure desired can be signalled from the control room.

The arrangement of burners on a boiler front plate is shown in Figure II.

For operating purposes, each oil burner is fitted with two valves and each boiler is fitted with a master control valve. In practice only the valve nearest the burner is used for lighting or shutting down a burner, the valve at the supply pipe being fitted as a precautionary measure. The master valve is open at all times, except in case of emergency.

Above each boiler room is a blower room in each of which is located a Sturtevant blower designed to deliver 25,000 cubic

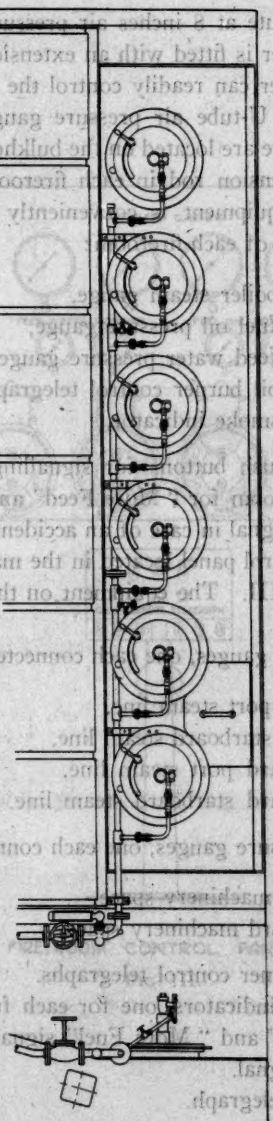


FIG. 11.—ARRANGEMENT OF B&W BURNERS ON BOILER FRONT

feet of air per minute at 8 inches air pressure. The throttle valve on each blower is fitted with an extension rod located so that the water-tender can readily control the blower from the fireroom floor. A U-tube air pressure gauge and a blower steam pressure gauge are located on the bulkhead alongside the blower throttle extension rod in each fireroom.

The following equipment is conveniently grouped on the bulkhead at one end of each fireroom:

One boiler steam gauge,
One fuel oil pressure gauge,
One feed water pressure gauge,
One oil burner control telegraph,
One smoke indicator.

There are also push buttons for signalling the machinery spaces and control room for "More Feed" and "More Fuel," and an emergency signal in case of an accident.

The fireroom control panel located in the main control room is shown in Figure III. The equipment on this panel consists of the following:

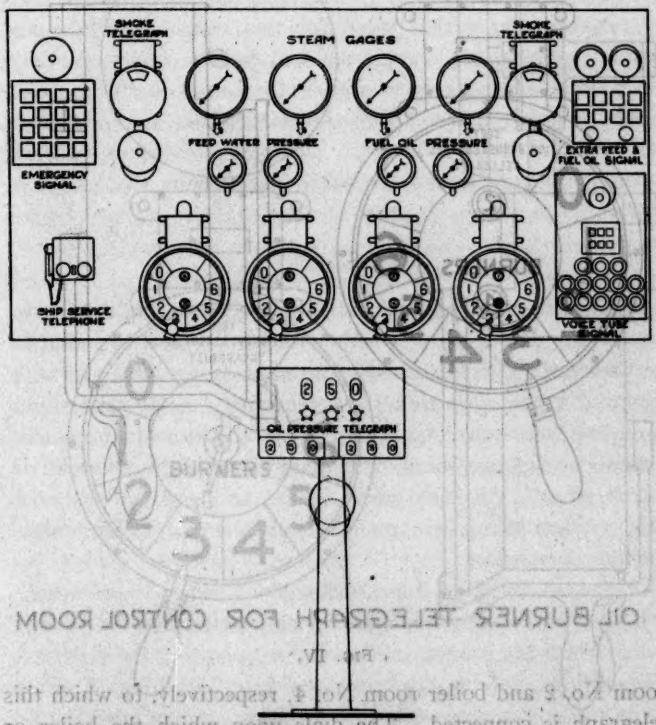
Four boiler steam gauges, one each connected to

After port steam line,
After starboard steam line,
Forward port steam line,
Forward starboard steam line.

Two fuel oil pressure gauges, one each connected to

After machinery space,
Forward machinery space.

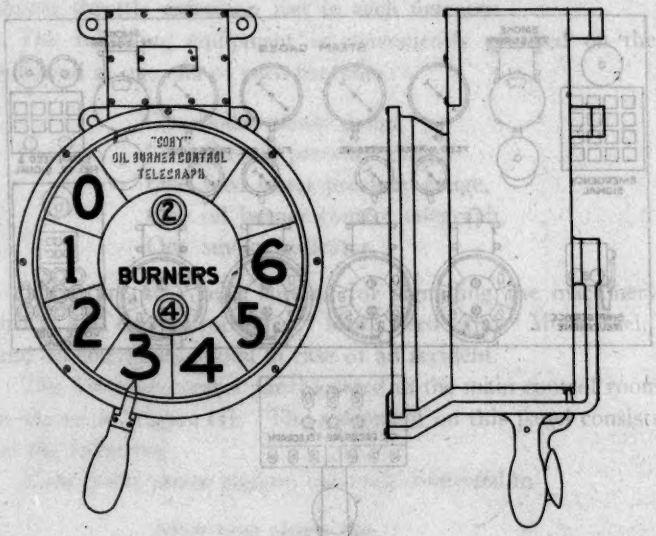
Four Cory oil burner control telegraphs.
Two Cory smoke indicators, one for each funnel.
One "More Feed" and "More Fuel" signal.
One emergency signal.
One oil pressure telegraph.



FIRE ROOM CONTROL PANEL.

FIG. III.

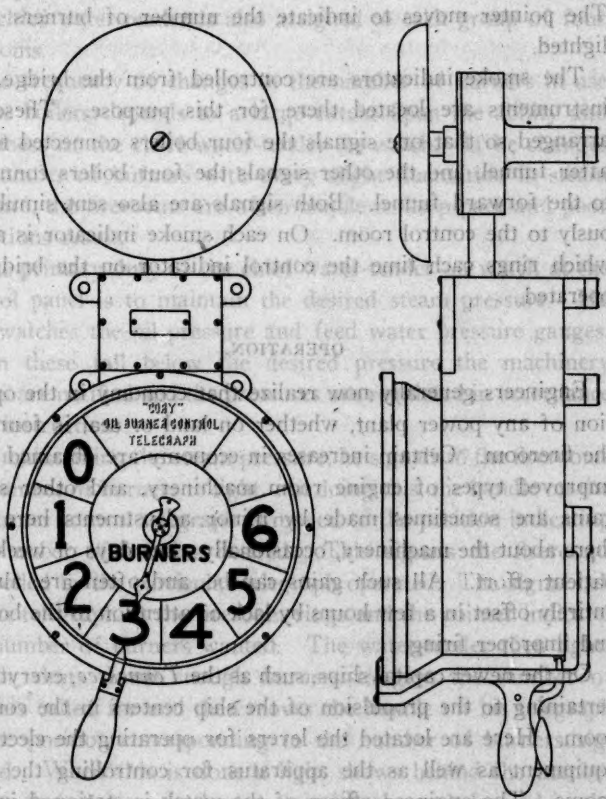
The type of oil burner telegraph located in the control room is shown in Figure IV. Each of these is arranged to control two firerooms simultaneously, thus: No. 1 and No. 3, No. 2 and No. 4, No. 5 and No. 7, No. 6 and No. 8. The figures on the dial, such as "2" and "4," Figure IV, represent boiler



OIL BURNER TELEGRAPH FOR CONTROL ROOM

FIG. IV.

room No. 2 and boiler room No. 4, respectively, to which this telegraph is connected. The dials upon which the boiler or fireroom numbers are painted are transparent and have an electric light placed at their rear inside the telegraph. These lights illuminate the fireroom number when that fireroom has answered the signal sent from the control room. The method of signalling will be described later.



OIL BURNER TELEGRAPH FOR FIREROOM

FIG. V.

The oil burner telegraph located in each fireroom is of the type shown in Figure V. This type is fitted with a signal bell, which rings whenever the control room telegraph is operated. The pointer moves to indicate the number of burners to be lighted.

The smoke indicators are controlled from the bridge, two instruments are located there for this purpose. These are arranged so that one signals the four boilers connected to the after funnel, and the other signals the four boilers connected to the forward funnel. Both signals are also sent simultaneously to the control room. On each smoke indicator is a bell which rings each time the control indicator on the bridge is operated.

OPERATION.

Engineers generally now realize that economy in the operation of any power plant, whether on land or sea, is found in the fireroom. Certain increases in economy are obtained with improved types of engine room machinery, and other slight gains are sometimes made by minor adjustments here and there about the machinery, occasionally after days or weeks of patient effort. All such gains can be, and often are, almost entirely offset in a few hours by lack of attention to the boilers and improper firing.

On the newer capital ships, such as the *Tennessee,* everything pertaining to the propulsion of the ship centers in the control room. Here are located the levers for operating the electrical equipment as well as the apparatus for controlling the firerooms. The engineer officer of the watch is stationed in the control room and keeps in constant touch with fireroom conditions. A chief water-tender operates the fireroom control.

Each fireroom is in charge of a water-tender. He is assisted by two firemen. One of the firemen looks after the burners and the other mans the J. V. telephone continuously. The J. V. telephone is also manned continuously in the control room

and the machinery spaces, so that any signals or instructions not possible to send by instruments can be immediately given by telephone.

A chief water-tender is also assigned to each group of four firerooms.

The frequency of changes in the number of burners in use on the boilers depends to a large extent upon the ability and steadiness of the chief water-tender on watch. If he changes the number of burners with every slight fluctuation in steam pressure, the firerooms are often unable to keep pace and poor conditions result.

The primary duty of the chief water-tender on watch at the control panel is to maintain the desired steam pressure. He also watches the oil pressure and feed water pressure gauges. When these fall below the desired pressure the machinery spaces are notified and the pressures are regulated in accordance with instructions.

In maintaining the steam pressure, he signals to the firerooms the number of burners to light. In doing this, the handle on the oil burner control telegraph is moved to the position indicating the number of burners desired. This signals the firerooms controlled by the particular telegraph operated. The signal bell on the telegraph in the fireroom rings and the pointer indicates the number of burners wanted. The water-tender then lights off or shuts down enough burners to give the number of burners called for and then moves the handle on his telegraph to the position corresponding to the number of burners signalled. When this is done a light shows behind the number of the fireroom on the burner control telegraph located in the control room. This procedure is repeated whenever the control room signals a change in the burners, so that the watch in the control room can tell by the signal lights on the control telegraphs just how many burners are in use on all boilers.

When a signal is sent from the control room to the fireroom for a certain number of burners, the water-tender on

watch speeds up or slows down the blower, according to the number of burners called for, and then lights off or shuts down the burners as required. The air pressure to be carried for any given number of burners and oil pressure is shown on a chart placed alongside the blower throttle extension handle. By reference to this chart the necessary air pressure is observed and the blower is regulated to give the desired pressure. The air pressure required for a certain number of burners varies with any change in the oil pressure. The oil pressure carried is shown by the oil pressure telegraph in the fireroom and can be readily checked by reference to the oil pressure gauge.

The chart used during the shake-down cruise and trials of the *Tennessee* is shown below;

FORCED DRAFT GUIDE.

Size Sprayer	Number Burners.	Draft in.	Blower Steam 1 Noz.	2 Noz.	3 Noz.	Oil Pressure.
42-20	6	4.0	220			200
	5	3.4	165			200
	4	3.1	130			200
	3	2.4	90			200
	2	2.0	70			200
42-20	6	4.8	250	150		250
	5	4.0	200	120		250
	4	3.5	150	95		250
	3	2.9	100	70		250
	2	2.4	75	55		250
42-20	6	5.6	190			300
	5	5.1	150			300
	4	4.2	110			300
	3	3.2	75			300
	2	2.8	60			300

The above chart was developed in the following manner: An electric light was hung opposite the peephole at the bot-

tom of the third pass on one side of a boiler. By observing this light from the peephole on the opposite side of the boiler the smoke condition was readily determined. It is well known that best combustion is indicated by a light haze. When the electric light is plainly visible across the boiler, yet *without a glare*, the smoke condition is right. This can be readily checked by observing the funnel from the deck.

In determining the various points on the chart the oil pressure was fixed first at 200 pounds. Two burners were lighted and the air pressure regulated until the smoke condition as observed through the peephole in the boiler was satisfactory. The air pressure was then noted. Later three burners were lighted, maintaining the oil pressure at 200 pounds and the same procedure was followed. After determining the correct air pressure also for four, five and six burners, the oil pressure was raised to 250 and 300 pounds successively and the air presure for the various number of burners was noted in a similar way. The results obtained on one boiler were checked against another boiler.

This chart was intended merely as a guide for the watertenders. It was found that, on some boilers, slight modifications were necessary in the air pressure shown on the chart. These modifications varied from time to time on the same boiler as small leaks developed in the fireroom, casing, etc. However, if for any given number of burners and oil pressure, the air pressure was carried at the point shown by the chart, it was found that the smoke condition was approximately correct. When frequent changes were made in the number of burners in use, as when maneuvering, this was sufficient. When the number of burners remained fairly constant, as when cruising, the slight correction in the air pressure could be made by observation and efficient combustion obtained.

Before obtaining the data necessary for the above chart it was, of course, necessary to thoroughly clean the atomizers and strainers, so that it would be certain that the oil pressure shown

on the gauge would be delivered to the atomizers. It might be said here that one of the two principal causes of lack of capacity and efficiency in oil burning is dirty atomizers. Too much attention cannot be given this point. Strainers must be kept clean and the slots in the sprayer plate or tip must also be kept clean and polished. Metal should never be used in polishing these surfaces, as even slight scratches will cause streaks in the oil spray. A tool made of lignum vitae proves very satisfactory for cleaning sprayer plates or tips.

The second main reason for lack of efficiency in oil burning is improper regulation of the air supply. Most of the firemen appreciate that there is a loss when smoke is produced, but few seem to realize the great loss that is obtained with the use of too much excess air, producing a clear funnel. One condition is as bad as the other and perhaps the efficiency is even less when running with a large amount of excess air than with a deficiency of air supply. After some practice an observer can tell when the right furnace conditions exist by watching the furnace through the peephole in the side casing. However, a simple indication of correct conditions is given by carrying a light haze at the funnel.

When the ship was operated under specified conditions during the shake-down cruise and trials, the number of boilers cut in was of course determined by the engineer officer. He also specified the oil pressure. The atomizers were fitted with the same size sprayer plates or tips for all conditions. When possible, and this was usually the case, all but one or two of the boilers were operated with a constant number of oil burners in use. The steam pressure was then regulated by varying the number of burners on the other one or two boilers. This made it possible to maintain fixed conditions on most of the boilers and increased their efficiency.

If the ship was operating under a wide range of speeds, all boilers were cut in, the oil pressure was fixed at 300 pounds and the steam pressure regulated by varying the num-

ber of burners in use. In fact, the ship was operated under all conditions from port load to full power, by merely changing the oil pressure successively from 200 to 250 and 300 pounds and by varying the number of burners. The time necessary to light five of the six burners, or shut down five of them, is but a few seconds.

It is understood that during the months that have elapsed since the trials of the *Tennessee* the ship has materially reduced her steam consumption at all loads and that the method of operating the firerooms is slightly different in certain details. However, the general method is essentially as herein described.

Certain other schemes have been suggested for controlling the firerooms and regulating all the boilers and burners simultaneously by operating one valve in the oil line. Of course, the air pressure would have to be regulated too, as well as other adjustments made. In this connection it should be remembered that the size of our present capital ships is such that regulation of machinery, to cause a change in speed, is not instantly effective in the headway of the ship. On such ships as the type herein described, the time necessary to regulate the burners in the firerooms is not appreciably, if at all, greater than the time necessary to regulate the electrical machinery to conform with the signal from the bridge.

The method of fireroom control used on the *Tennessee* is also in use on the new battleship *Maryland*. On both ships the scheme has proved entirely satisfactory and very flexible.

When operating at full power it was found that the boilers that were farthest away from the turbine throttles tended to lift their safety valves even though the line pressure was below the safety valve setting. This was readily overcome by setting up five pounds on these safety valves.

An improvement in the fireroom control apparatus adopted on the *Maryland* consists of arranging an air pressure gauge so that the control room can observe the air pressure in any fireroom at any time. If the smoke indicator shows that some

fireroom is smoking, the man on watch in the control room can easily determine what fireroom or firerooms it is and give orders to remedy it.

There are a few other improvements that seem desirable to install in the control room. For instance, a cut-out switch should be attached to the oil burner control telegraph so that, if desired, only one fireroom at a time could be signalled. Occasionally the steam pressure varies so little that a change in the number of burners in use on two boilers is more than is needed. On larger ships, such as those with sixteen boilers, it would be advisable to provide a system of easily operated switches that would enable one oil-burner control telegraph to operate one boiler; two, three or four boilers; or various combinations of boilers.

Practical distant reading thermometers are now available and it would be comparatively easy to install thermometers of this type that would indicate the fuel oil and feed water temperature on the fireroom control panel. The necessity for keeping these temperatures constant needs no emphasis here. Nevertheless, trouble is often experienced with fluctuating temperatures and it would be well to have thermometers indicating such temperatures located where they would come under the constant observation of the control room.

It is believed that the addition of the above noted apparatus will improve the efficiency of these ships. However, the extreme flexibility and simplicity of operation of the fireroom control on the *Tennessee* as described herein, has been amply demonstrated in service.

THE DYSON BOILER.

By Lieut. Comdr. H. G. Cooper, U. S. N., M. S., Member.

In August, 1919, the Bureau of Engineering submitted to the Navy Department the plans for a boiler which had been designed by Rear Admiral C. W. Dyson, U. S. N. The Bureau stated that it believed that this boiler would prove a particularly satisfactory steam generator for capital ships, and recommended that, from the plans submitted, one boiler be constructed for experimental purposes. The Navy Department authorized the construction of such a boiler, and awarded the contract for it to the Newport News Shipbuilding and Dry Dock Company.

The boiler was duly constructed in exact accordance with the original plans, and, in January, 1921, was delivered to the Philadelphia Navy Yard where it was set up at the Fuel Oil Testing Plant. After certain preliminary tests had been run to ascertain that the boiler was in proper condition, a Board was appointed to conduct official tests and to report on the results. This article is based on the results of these tests as embodied in the official report of the Board.

DESCRIPTION OF THE BOILER.

The boiler was designed to meet the requirements for a steam generator for high-powered vessels of the Navy, these requirements being:

1. High efficiency. The specifications for this class of boiler require that, when burning oil at the rates of 0.25, 0.5, 0.75 and 1 pound of oil per square foot of heating surface per hour on the basis of oil having 19,500 B. T. U's. per pound,

the evaporative efficiencies from and at 212 degrees F. shall be not less than 16.2, 15.75, 15.1 and 14.5 pounds of water per hour, respectively, per pound of oil.

2. Great heating surface compared to the weight and volume of the boiler,

3. Ability to stand forcing to the extent of at least 1½ pounds of oil per square foot of heating surface.

4. Furnace conditions that will permit smokeless operation at high steaming rates, and that will meet the specified requirements that "the maximum rate of combustion shall not exceed 10.8 pounds of oil per cubic foot of furnace volume per hour."

5. Superheaters that will provide a reasonably high degree of superheat, and through which the steam velocity and the pressure drop must be low, and which will meet the requirements of the specifications, as follows: "The superheaters shall be so located as to permit of easy access and removal for examination, renewal, or repairs. The superheating tubes shall be so located among the generating tubes that the superheater tubes shall in no place be closer to the furnace than one-half the thickness of the generator tube nest, and so arranged that the entire products of combustion must pass across the superheating surface."

6. The absence of plate gas baffles, as these are prohibited by the specifications.

The boiler is of the Express type, and consists of one upper or steam drum, connected by two nests of tubes to two lower, or water drums. These tubes are straight except at the ends where they are bent to enter the drums normally. The two inner rows of tubes of each nest are 1⅜ inches in external diameter by 120 mils thick, the remaining tubes are 1 inch in external diameter by 109 mils thick. In the outer half of each nest of tubes every third row of tubes in the direction of the axis of the drums is omitted to provide a space to be occupied by the superheater tubes. On the uptake sides of the

generating tubes are located the two superheating units, one unit on each side, each unit consisting of two 15-inch drums connected by curved tubes 1 inch in external diameter by 109 mils thick. The shape of the superheater tubes is such that when the superheaters are in place the superheater tubes occupy the spaces between the outer generating tubes in such a manner that the entire products of combustion must pass across the superheating surface. As originally designed, the only gas baffling consisted of a baffle wall about 15 inches wide on each superheating unit, the upper edge of the wall touching the upper superheater drum and resting on the outer row of superheater tubes. Retarders are installed in the outer three rows of tubes of each superheating unit. An idea of the appearance of the boiler can be obtained from Plates 1 and 2.

The principal dimensions of the boiler are:

Generating surface, square feet..... 6,170
Superheating surface, square feet..... 1,016
Total heating surface, square feet..... 7,186
Working steam pressure, pounds..... 280
Furnace volume, cubic feet..... 654
Estimated superheat at 1 pound oil per square foot of water heating surface 60 degrees to 80 degrees F.
Generating tubes, 1⅜ inches outside diameter by 120 mils thick, approximate length 9 feet 3 inches, number 174.
Generating tubes, 1-inch outside diameter by 109 mils thick, approximate mean length 8 feet 6 inches, number 2,500.
Superheating tubes, 1-inch outside diameter by 109 mils thick, maximum length 7 feet 4⅛ inches, minimum length 4 feet 8⅝ inches, number 780.
Steam drum, 48 inches inside diameter, 12 feet 1¼ inches long, number 1.
Water drum, 15 inches inside diameter, 11 feet 11½ inches long, number 2.
Superheater drums, 15 inches inside diameter, 11 feet 9¾ inches long, number 4.

OBJECT OF THE TESTS.

1. To ascertain at various rates of combustion, the evaporative efficiency of the boiler and furnace as designed.

2. To determine the value of this boiler as compared with other boilers of the same type, as a steam generator for battleships, battle cruisers and scout cruisers.

3. To determine, as far as possible, alterations in design and construction that will improve the efficiency of the boiler and its fitness for Naval use.

GENERAL DESCRIPTION OF TESTS.

All tests were conducted with the boiler equipped with nine Bureau of Engineering Standard Natural-Forced Draft Air Registers and Bureau of Engineering Standard Fuel Oil Atomizers. Plates 1 and 2 show the arrangement of the registers in the boiler front. The first series of tests (Nos. 38–1 to 38–5) were made with the boiler arranged in every way exactly as designed, as shown on Plate 3. The remaining tests were made with various arrangements of gas baffling, as shown on Plates 4, 5 and 6. The size and position of the holes in the distributor pipes in the lower superheater drums were changes between tests as noted below. After test No. 38–21 and before test No. 38–22, moulded refractory furnace opening rings were substituted for the plastic clay rings used in the preceding tests. No other changes than these were made to the boiler during the course of the tests.

The official tests were started on March 1, 1921, and continued thereafter until completion. The dates on which the various tests were run are shown on the summary sheets, Plates 7, 8 and 9.

RESULTS OF THE TEST.

The summary sheets, Plates 7, 8 and 9, and the curves plotted therefrom, Plates 10, 11 and 12, show the results of the tests.

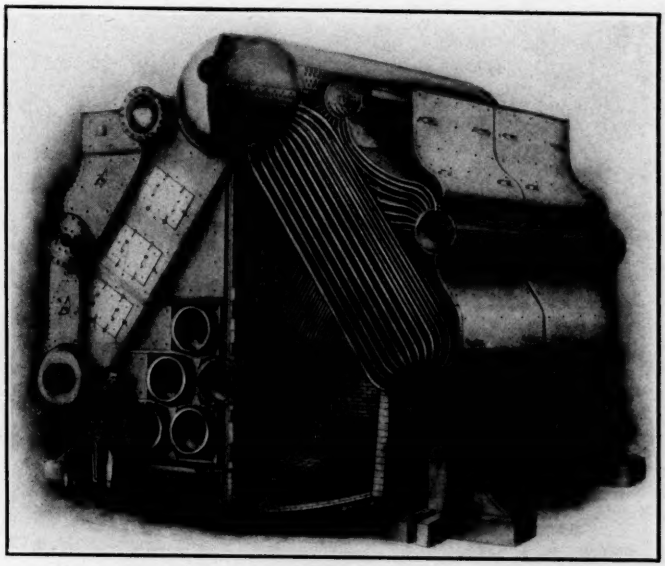
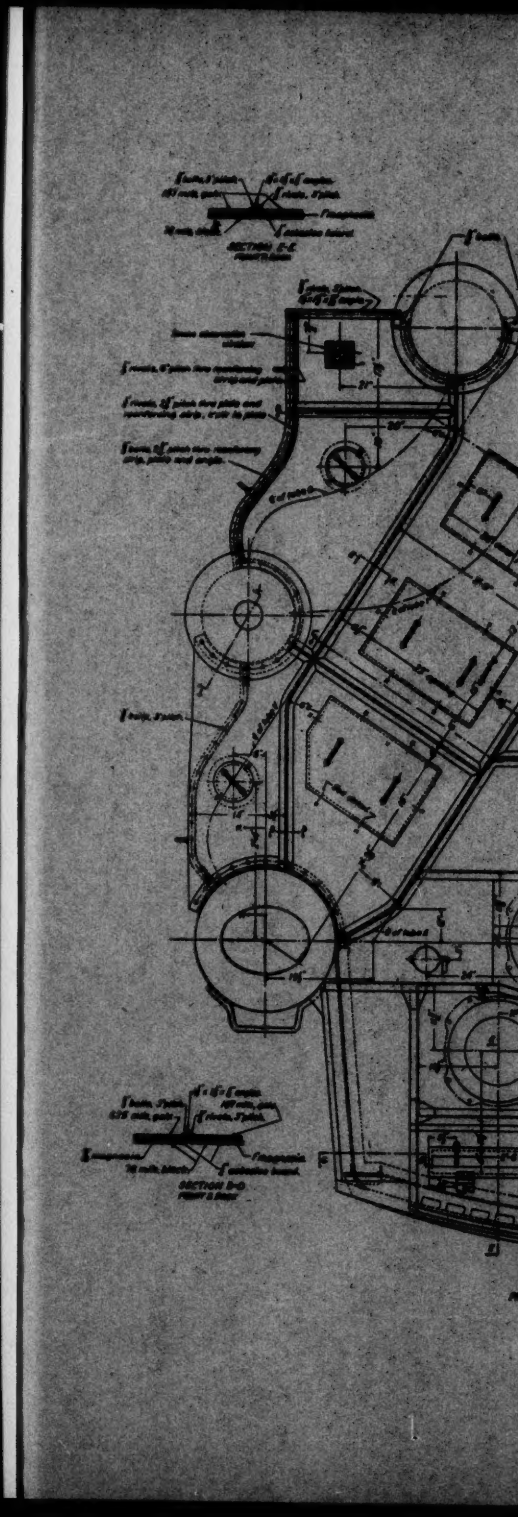


PLATE 1.



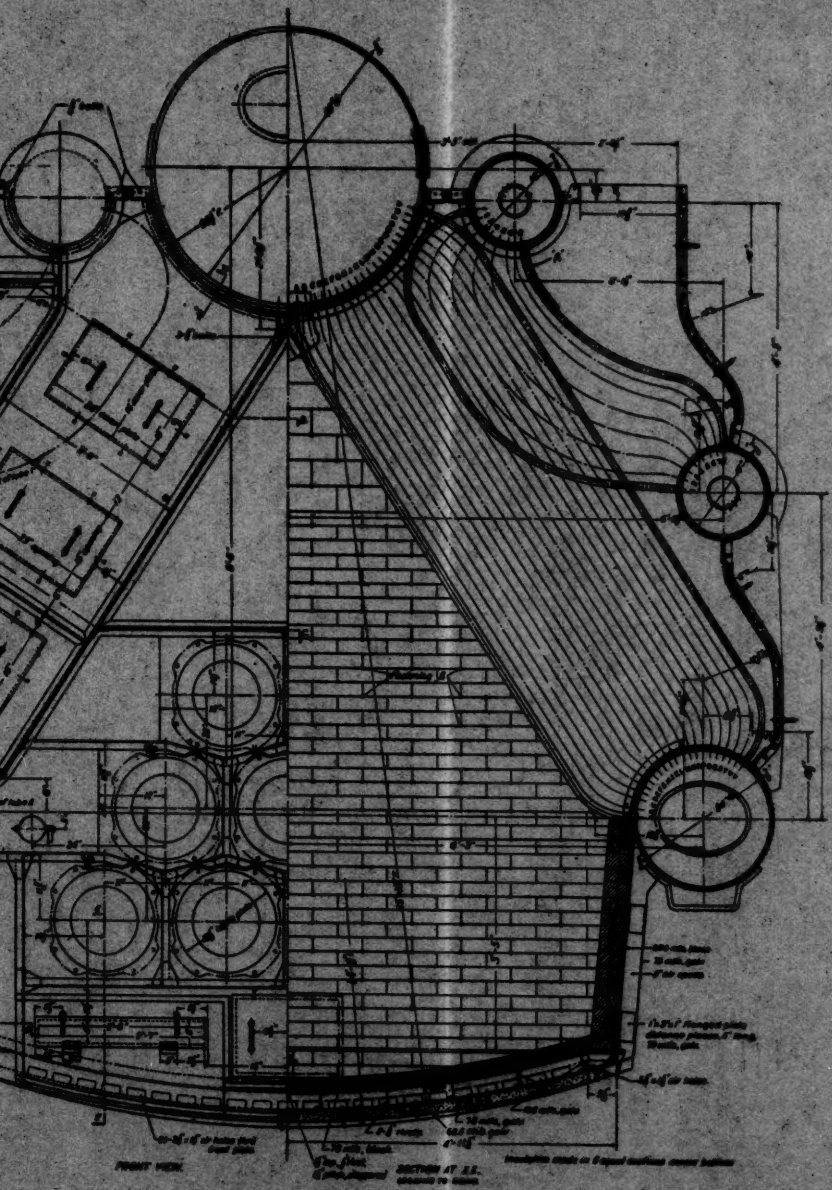


PLATE 2.

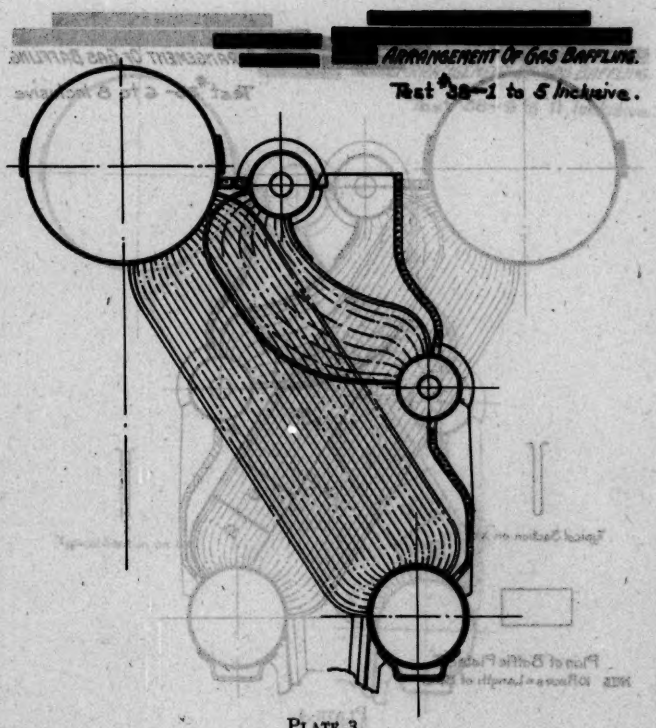


PLATE 3.

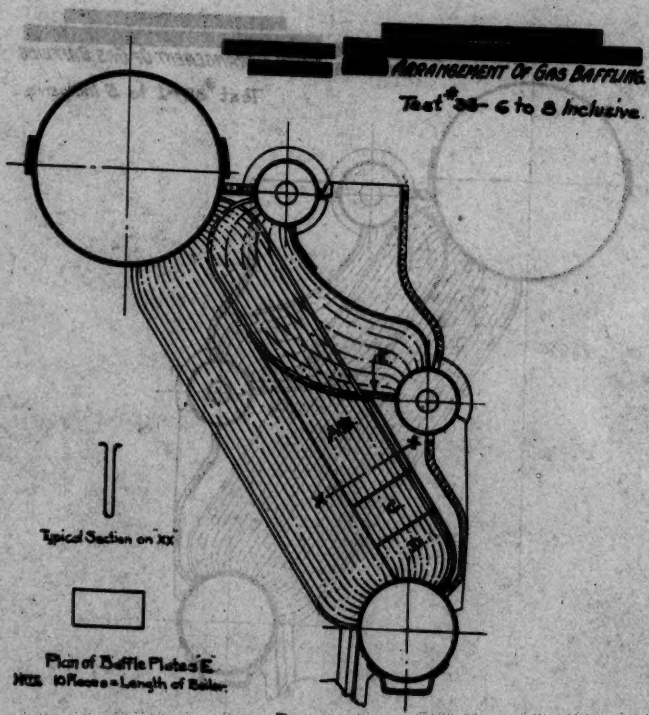

Arrangement of Gas Baffling.
Test #38 - 6 to 8 Inclusive.
Typical Section on XX
Plan of Baffle Plates E.
NOTE 10 Pieces = Length of Boiler.

PLATE 4.

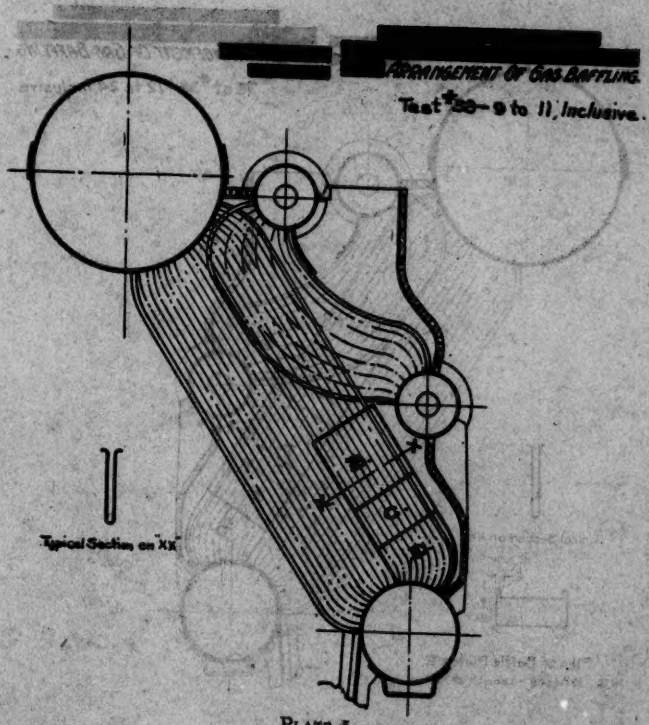

Arrangement of Gas Baffling.
Test No. 9 to 11, Inclusive.
Typical Section on XX

PLATE 5.

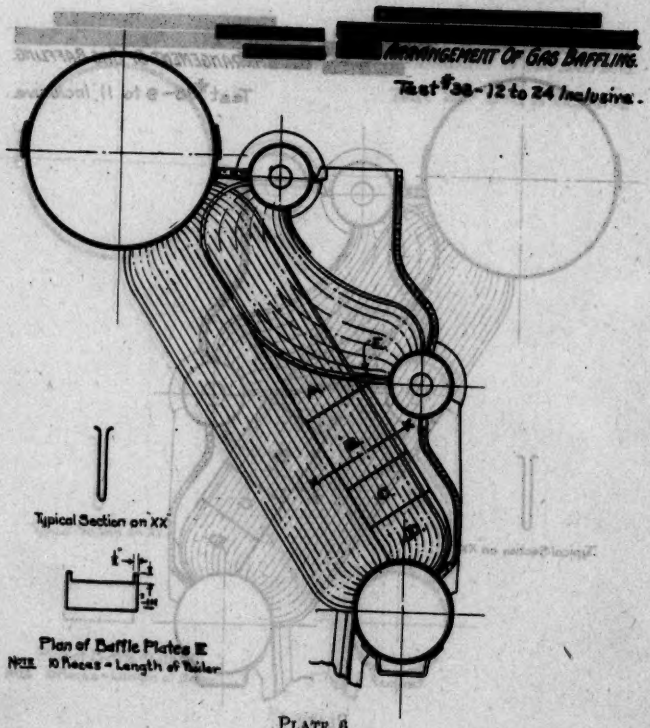

ARRANGEMENT OF GAS BAFFLING.
Test Nos 12 to 24 Inclusive.
Typical Section on XX
Plan of Baffle Plates E
Nos 10 Recess = Length of Boiler

PLATE 6.

FUEL OIL TESTING PLANT

Water heating surface 6170 Sq.ft
Superheating Surface 1016 Sq.ft
Total heating surface 7186 Sq.ft
Furnace Volume 634 Cu.ft

EVAPORATIVE TEST ON DIXON SUPERHEATER, ECONOMIZER-BOILER
EQUIPPED WITH EM STR. INDUCED FORCED DRAFT AIR REGULATORS
AND EM STR. FUEL OIL ATTOMIZERS.

Number of tubes 174 - 2500 - 100 Superheater
Size of Tubes 1⅞"OD - 1/2" - 1/2"
Diameter of steam drum 48"ID
Working pressure 280 lbs. per sq. inch gauge.

1	Test Number								
2	Date								
3	Barometer - Inches of mercury								
4	Duration of test								
4a	Number of burners used								
5	Size of atomizers								
6	Arrangement of burners used								
6a	Type of air regulators used								
6	Aggregate opening of air regulators - Sq inches								
	STEAM OBSERVATIONS								
7	Steam pressure as drawn - Lbs. per sq. inch gauge								
8	Steam pressure before boiler stop - lbs. per sq. inch gauge								
9	Superheater temperature - °F								
10	Calorimeter temperature (vapor temperature) °F								
11	Calorimeter (inlet atmospheric) Lbs. per sq inch gauge								
12	Feed water temperature at boiler - °F								
13	Water actually evaporated - Lbs. per hour								
	FUEL OBSERVATIONS								
14	Oil burned - Lbs. per hour								
15	Oil pressure at burners - Lbs. per sq inch gage								
16	Oil temperature at burners - °F								
	AIR OBSERVATIONS								
17	Dry bulb temperature - °F								
18	Wet bulb temperature - °F								
19	Firebox temperature - °F								
19a	Stack temperature - °F								
20	Furnace temperature (by optical pyrometer) - °F								
21	Furnace pressure - Inches of water								
22	Air pressure in burners - Inches of water								
23	Air pressure at base of stack - Inches of water								
24	Excess by Ringelmann scale								
	FLUE GAS ANALYSIS	1 2 3	1 2 3	2 3	1 2 3	2 3	1 2 3	2 3	1 2 3 1 2 3
25	Sampling tube number								
26	Carbon dioxide - %								
27	Oxygen - %								
28	Carbon monoxide - %								
29	Nitrogen by difference - %								
	ULTIMATE ANALYSIS & PHYSICAL TESTS OF FUEL								
30	Carbon - %								
31	Hydrogen - %								
32	Sulphur - %								
33	Nitrogen - %								
34	Oxygen and undetermined - %								
35	Gravity - Baumé								
36	Gravity - Specific								
37	Flash point - °F								
37a	Viscosity temperature - °F								
38	Viscosity - Engler								
39	Water & sediment - %								
40	Calorific value - B.T.U. per lb.								
	STEAM RESULTS								
41	Superheat - °F								
42	Quality of steam (steady state) - %								
43	Quality of steam (average) - %								

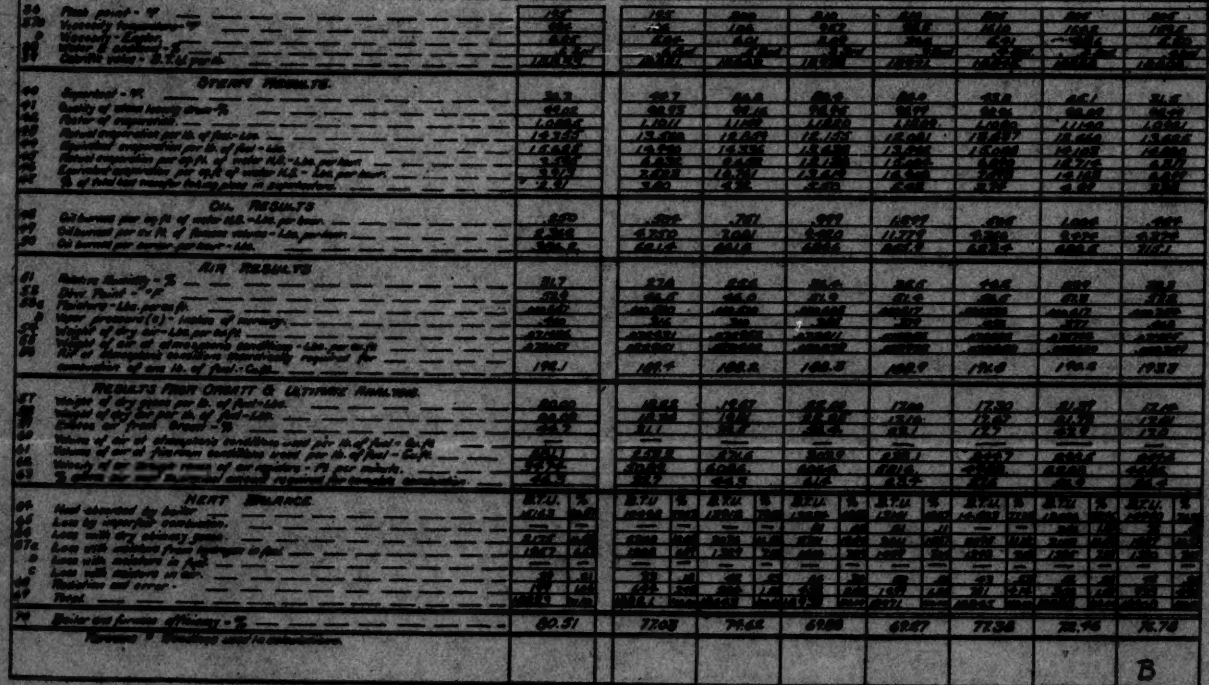


PLATE 7.

FUEL OIL TESTING PLANT

EVAPORATIVE TEST ON DIXON EXPERIMENTAL EVAPORATOR
EQUIPPED WITH OIL STR. FURNACE, FORCED DRAFT AIR REGISTERS
AND OIL STR. FUEL OIL ATOMIZERS.

Water heating surface 6170 Sq. Ft.
Superheating surface 1016 " "
Total heating surface 7186 " "
Furnace Volume 1408 Cu. Ft.

Number of tubes 178 - 2.50 - 20.0 Grade A
2) No. of tubes 178 - 2.50 - 20.0 "
3) Number of tubes 32 - 2.50 - 20.0 "
Working pressure 200 Lbs. per sq. inch.

1	Test Number	1	2	3	4	5	6	7	8
2	Date	6-7-04	6-7-04	6-7-04	6-7-04	6-7-04	6-7-04	6-7-04	6-7-04
3	Barometer - Inches of mercury	29.91	29.91	29.91	29.91	29.91	29.91	29.91	29.91
4	Duration of Test	8 hr.	8 hr.	8 hr.	8 hr.	8 hr.	8 hr.	8 hr.	8 hr.
4a	Number of burners used	4	4	4	4	4	4	4	4
5	Size of atomizers	5-5-5-5	5-5-5-5	5-5-5-5	5-5-5-5	5-5-5-5	5-5-5-5	5-5-5-5	5-5-5-5
6	Arrangement of burners used	6-7-04	6-7-04	All.	6-7-04	All.	All.	6-7-04	6-7-04
6a	Type of air registers used	Adj. Reg.	Adj. Reg.	Adj. Reg.	Adj. Reg.	Adj. Reg.	Adj. Reg.	Adj. Reg.	Adj. Reg.
6b	Approx. opening of air registers - Sq. inches	324	324	324	324	324	324	324	324
	STEAM OBSERVATIONS								
7	Steam pressure by chart - Lbs. per sq. inch gauge	224.6	224.3	224.2	224.2	224.4	224.6	224.6	224.7
8	Steam pressure before boiler stop - Lbs. per sq. inch gauge	218.2	218.2	218.2	218.1	218.3	218.3	218.2	218.2
9	Superheat temperature - °F	338.0	337.4	338.6	338.4	338.4	338.1	338.1	338.3
10	Calorimeter temperature (Loss corrected) °F	340.0	340.2	340.4	340.4	340.4	340.3	340.3	340.4
11	Calorimeter (vacuum or pressure) Lbs. per sq. inch gauge	24.5	24.5	24.5	24.5	24.5	24.5	24.5	24.5
12	Feed water temperature by meter - °F	60.0	60.0	60.0	60.0	60.0	60.0	60.0	60.0
13	Water actually evaporated - Lbs. per hour	20,000	20,000	20,000	20,000	20,000	20,000	20,000	20,000
	FUEL OBSERVATIONS								
14	Oil burned - Lbs. per hour	2,082.5	2,110.6	2,100.0	2,092.0	2,090.0	1,999.5	1,991.7	1,991.7
15	Oil pressure at burners - Lbs. per sq. inch gauge	200.1	199.7	199.8	199.8	200.1	200.0	200.0	200.0
16	Oil temperature at burners - °F	204.2	204.0	204.2	204.3	204.5	204.5	204.5	204.5
	AIR OBSERVATIONS								
17	Dry bulb temperature - °F	72.0	72.1	72.2	72.5	72.1	72.1	72.1	72.1
18	Wet bulb temperature - °F	59.0	59.2	59.2	59.3	59.2	59.2	59.2	59.2
19	Relative humidity - %	57.0	57.0	57.0	57.0	57.0	57.0	57.0	57.0
20	Humidity - Grains of water per lb. of dry air	61.0	61.0	61.0	61.0	61.0	61.0	61.0	61.0
20a	Humidity - Lbs. of water per lb. of dry air	0.0087	0.0087	0.0087	0.0087	0.0087	0.0087	0.0087	0.0087
21	Furnace temperature (By optical pyrometer) - °F	2,400	2,400	2,400	2,400	2,400	2,400	2,400	2,400
22	Furnace pressure - Inches of water	+0.10	+0.10	+0.10	+0.10	+0.10	+0.10	+0.10	+0.10
23	Air pressure at burner - Inches of water	+2.50	+2.50	+2.50	+2.50	+2.50	+2.50	+2.50	+2.50
24	Air pressure at loss of draft - Inches of water	-0.05	-0.05	-0.05	-0.05	-0.05	-0.05	-0.05	-0.05
25	Excess air - Percentage over	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5
	FLUE GAS ANALYSIS								
26	Sample from number	2	2	2	2	2	2	2	2
27	Carbon dioxide - CO_2	12.0	12.0	12.0	12.0	12.0	12.0	12.0	12.0
28	Oxygen - O_2	6.0	6.0	6.0	6.0	6.0	6.0	6.0	6.0
29	Carbon monoxide - CO	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
30	Nitrogen by difference - N_2	82.0	82.0	82.0	82.0	82.0	82.0	82.0	82.0
	ULTIMATE ANALYSIS & PHYSICAL TESTS OF FUEL								
31	Carbon - C	85.67	85.73	85.73	85.68	85.68	85.73	85.68	85.68
32	Hydrogen - H	11.53	11.51	11.51	11.53	11.53	11.51	11.53	11.53
33	Sulphur - S	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7
34	Oxygen - O	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0
35	Water and impurities - W	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0
36	Heating value - B.T.U. per lb.	18,450	18,450	18,450	18,450	18,450	18,450	18,450	18,450
37	Gravity - Specific	0.965	0.964	0.964	0.965	0.965	0.965	0.965	0.965
38	Flash point - °F	300°	300°	300°	300°	300°	300°	300°	300°
39	Viscosity - Engler	3.5	3.5	3.5	3.5	3.5	3.5	3.5	3.5
40	Viscosity - Saybolt	200"	200"	200"	200"	200"	200"	200"	200"
41	Water vapor - Lbs. per lb.	0.0087	0.0087	0.0087	0.0087	0.0087	0.0087	0.0087	0.0087
	STEAM RESULTS								
42	Quality - °F	338	338	338	338	338	338	338	338
43	Quality of steam leaving drum - %	99.0	99.0	99.0	99.0	99.0	99.0	99.0	99.0
44	Heat of evaporation	1,000.0	1,000.0	1,000.0	1,000.0	1,000.0	1,000.0	1,000.0	1,000.0
45	Heat of superheat	13.36	13.37	13.38	13.38	13.38	13.37	13.37	13.37
46	Equivalent evaporation from and at 212°	13,360	13,370	13,380	13,380	13,380	13,370	13,370	13,370
47	Boiler horsepower developed	1,336	1,337	1,338	1,338	1,338	1,337	1,337	1,337

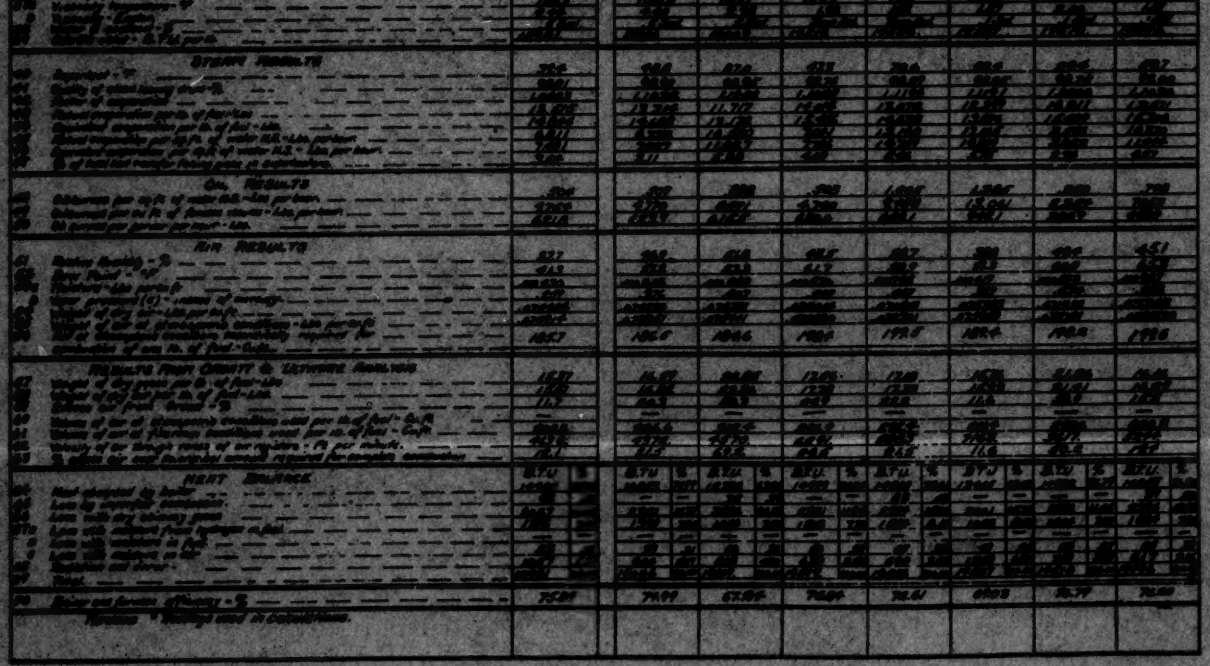


PLATE 8.

FUEL OIL TESTING PLANT

EVAPORATIVE TEST ON DIXON EXPERIMENTAL BOILER EQUIPPED WITH DELSTE INTERNAL FORCED DRAFT ATTACHMENTS AND DELSTE FUEL OIL ATOMIZERS.

Water heating surface 6170 Sq. Ft.
Total heating surface 7016 Sq. Ft.
Grate surface 77.62 Sq. Ft.
Furnace Volume 604 Cu. Ft.

Number of tubes 174 - 2500-7800 Standard
Size of tubes 1 3/4 OD - 10 B.W.G. - 10 ft. 0 in. long
Diameter of drum 84 in. I.D.
Working pressure 200 lbs. per sq. inch gauge.

1	Test Number								
2	Date								
3	Duration - hours of duration								
4	Duration of test								
4a	Number of burners used								
5	Size of atomizers								
6	Arrangement of burners used								
6a	Type of air registers used								
6b	Approximate opening of air registers - Sq. inches								
	STEAM OBSERVATIONS.								
7	Steam pressure as drawn - Lbs. per sq. inch gauge								
8	Steam pressure behind boiler gate - Lbs. per sq. inch gauge								
9	Superheater temperature - °F.								
10	Calorimeter temperature (hood connection) °F.								
11	Calorimeter (orifice drop pressure) Lbs. per sq. inch gauge								
12	Flue gas temperature at boiler - °F.								
13	Water actually evaporated - Lbs. per hour								
	FUEL OBSERVATIONS.								
14	Oil burned - Lbs. per hour								
15	Oil pressure at burners - Lbs. per sq. inch gauge								
16	Oil temperature at burners - °F.								
	AIR OBSERVATIONS.								
17	Dry bulb temperature - °F.								
18	Wet bulb temperature - °F.								
18a	Humidity								
18b	Stack temperature - °F.								
19	Furnace temperature (at optical pyrometer) - °F.								
20	Furnace pressure - Ins. of water								
21	Air pressure in boiler - Ins. of water								
22	Air pressure at base - Ins. of water								
23	Weight of Combustion used								
24	Weight of Combustion ash								
	FLUE GAS ANALYSIS.								
25	Carbon dioxide - %								
26	Oxygen - %								
27	Carbon monoxide - %								
28	Nitrogen by difference - %								
	ULTIMATE ANALYSIS & PHYSICAL TESTS OF FUEL.								
29	Carbon - %								
30	Hydrogen - %								
31	Nitrogen - %								
32	Oxygen - %								
33	Sulphur and constituents - %								
34	Ash - %								
35	Moisture - %								
36	Heat value - B.T.U. per lb.								
37	Flash point - °F.								
37a	Viscosity Saybolt - °F.								
38	Specific gravity - °F.								
39	Weight of oil - Lbs. per gal.								
40	Weight of oil - Lbs. per cu. ft.								
	ENERGY RESULTS.								
41	Evaporation - °F.								
42	Equiv. of water heated from - °F.								
43	Equiv. of water evaporated from - °F.								
44	Water evaporated per lb. of fuel as fired								
45	Water evaporated per lb. of dry fuel								
46	Water evaporated per lb. of combustible								
47	Boiler horsepower developed								
48	Efficiency of boiler and grate								
49	Efficiency of boiler, grate and superheater								
50	Heat absorbed per lb. of fuel as fired								
51	Heat absorbed per lb. of dry fuel								
52	Heat absorbed per lb. of combustible								
53	Heat absorbed per hour								
54	Heat absorbed per hour per sq. ft. of heating surface								
55	Heat absorbed per hour per cu. ft. of furnace								
56	Combustible per hour per sq. ft. of grate surface								
57	Combustible per hour per cu. ft. of furnace								
58	Ratio of heating surface to grate surface								
59	Ratio of furnace volume to heating surface								
60	Ratio of furnace volume to grate surface								

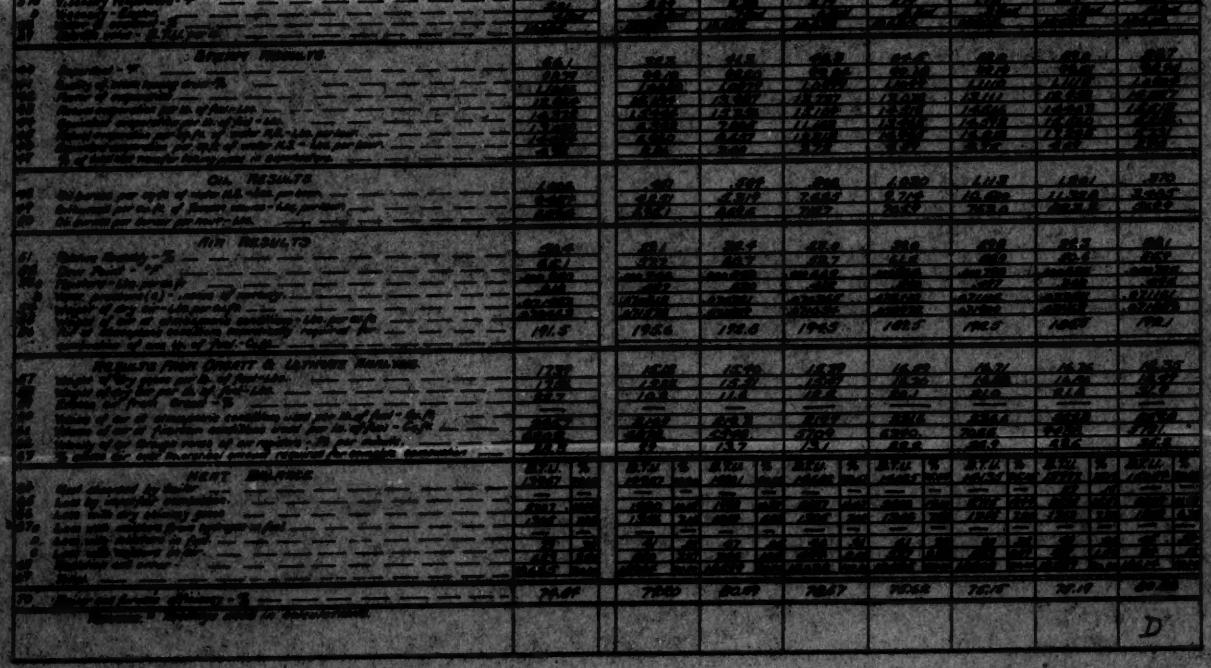


PLATE 9.

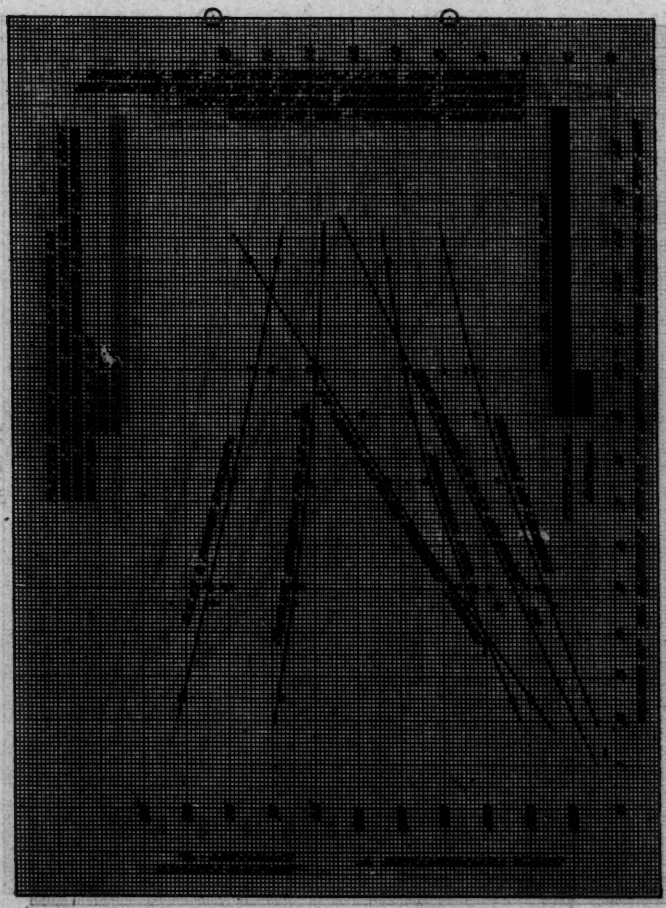


PLATE 10.

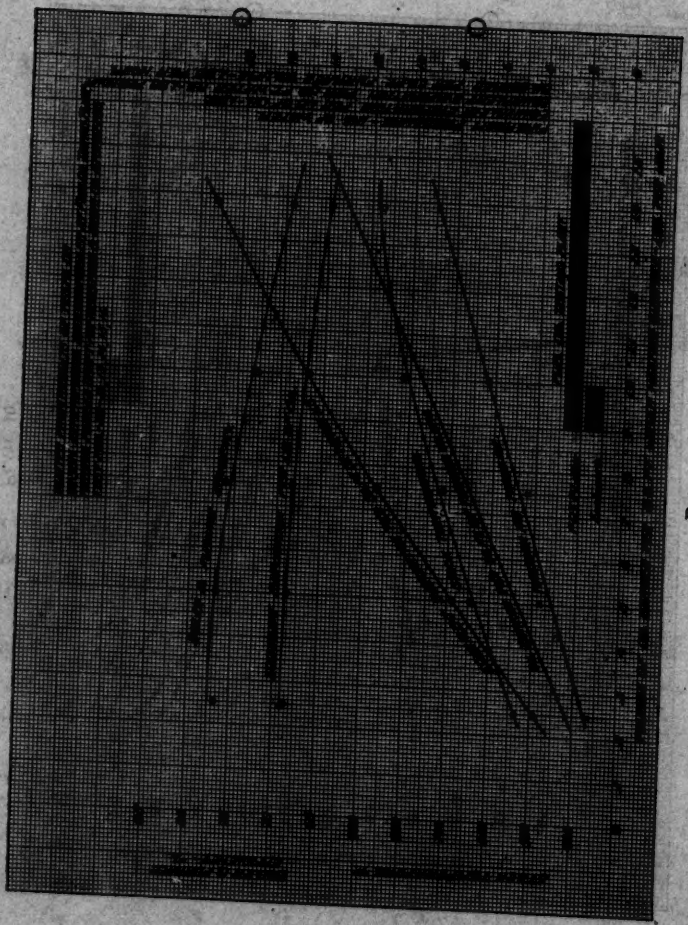


PLATE 11.

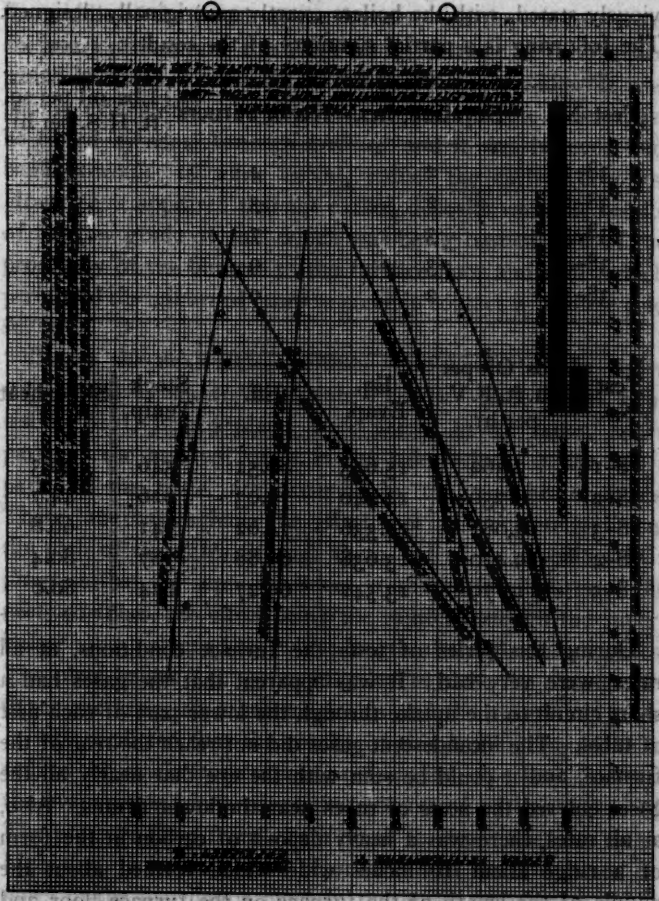


PLATE 12.

The first series of tests, Nos. 38–1 to 38–5, were conducted with gas baffling arranged as shown on Plate 3, and, as previously stated, with the boilers exactly as originally designed. The following is a partial summary of the results:

Test No.	Duration, hrs.	No. of burners	Lbs. oil per sq. ft. H.S. per hr.
38–1	2	4	.250
38–2	2	3	.504
38–3	3	7	.751
38–4	3	9	.999
38–5	2	9	1.249

Test No.	Lbs. Oil per Cu. ft. F.V. per hr.	Eq. Evap.	Eff.	Stack Temp.	Superheat
38–1	2.362	15.625	80.51	546	30.3
38–2	4.750	14.940	77.03	626	44.7
38–3	7.081	14.336	74.62	741	66.8
38–4	9.420	13.638	69.88	787	80.4
38–5	11.779	13.542	69.27	824	80.0

During this series of tests the furnace conditions, at all rates, were very bad. It was apparent that the gases had a short circuit to the uptake through the lower part of each nest of tubes. The incandescent gases did not reach the rear of the furnace, and it could be seen with the eye that nearly all the gases entered the nests of tubes near the lower drums. Also, at all rates, there was a heavy deposit of carbon in the form of a ridge about 10 inches wide which formed about the middle of the length of the furnace on the furnace floor and extended up each side of the furnace for about half the length of the inner rows of tubes. From time to time the carbon deposited on the tubes would break up and fall to the furnace floor in large solid lumps. The deposits, while forming on

the tubes, had somewhat the appearance of stalactites. No other carbon was deposited than this ridge which formed about the middle of the length of the furnace. At the end of five or six hours of steaming this ridge along the floor of the furnace would be about 8 inches in height by 10 inches wide. It was plainly evident that the short circuit path of the gases must be closed by some form of baffling, and it was believed that baffling would, to some extent, reduce the carbon deposits. However, it appeared that the carbon deposits were the results of the low grade of fuel which was being burned, and of the proximity of the burners to the furnace floor and walls which permitted the unburned gases to come in contact with these cold surfaces.

Before the next series of tests were conducted the boiler was baffled as shown on sketch, Plate 4. Lane baffles, consisting of U-shaped plates were placed in the outer, lower parts of the nests of tubes in the lanes which, in the upper parts of the nests, are occupied by the superheater tubes. These lane baffles have the effect, not of completely closing the path of the gases through them, but of so impeding their progress that the same resistance is offered in all parts of the nests of tubes, the effect being the same as though the superheater tubes extended the whole length of the generating tubes. The plate baffles "E" were installed to close the passage between the lower superheaters drums and the generating tubes. It was predicted that the results of this baffling would be to reduce the stack temperatures, reduce the superheat, increase the evaporation, and reduce the amount of carbon deposited. The pressure drop from steam drum to steam line was rather high, and, to improve this, before the next series of tests, the holes in the distributor pipes in the lower superheater drums were enlarged from ¼ inch to 9/32 inch, and the pipes were rotated 90 degrees so that the holes faced the inner walls of the drums instead of the lower walls. The following is a partial summary of the results of the tests with this arrangement:

Test No.	Duration, hrs.	No. of burners	Lbs. oil per sq. ft. H.S. per hr.
38–6	3	5	.505
38–7	2	9	1.004
38–8	2	4	.464

Test No.	Lbs. Oil per Cu. ft. F. V. per hr.	Eq. Evap.	Eff.	Stack Temp.	Superheat
38–6	4.766	15.028	77.38	594	43.8
38–7	9.474	14.103	72.46	696	65.1
38–8	4.374	14.880	76.78	572	31.5

As predicted, the stack temperatures during these tests were reduced to a satisfactory point, the superheat fell off materially, and the efficiency increased, but to a less degree than was expected. It was apparent after these three tests that further changes in the gas baffling would have to be made, so no further tests were run with this arrangement. During these tests the furnace conditions were much improved over what they were during the first series, but the carbon deposit, although slightly less than previously, was still very bad. It was believed that this condition was due to the quality of the oil being burned. The inferiority and unsuitableness of the oil was evidenced by the fact that sparks of incandescent solids could be observed impinging on the rear wall of the furnace.

Before the next series of tests the baffling was altered as shown on sketch, Plate 5. The plate baffles between the lower superheat drums and the generating tubes were removed completely, and the upper part of the lane baffles were removed. It was expected that this would improve the superheat without lowering the efficiency, and at the same time keep the stack temperatures reasonably low. These tests were conducted with this arrangement, with the following results.

Test No.	Duration, hrs.	No. of burners	Lbs. oil per sq. ft. H.S. per hr.
38–9	2½	5	.504
38–10	2	5	.507
38–11	2	9	.898

Test No.	Lbs. Oil per Cu. ft. F. V. per hr.	Eq. Evap.	Eff.	Stack Temp.	Superheat
38–9	4.754	14.844	75.89	633	39.4
38–10	4.770	14.688	74.99	670	48.8
38–11	8.470	13.172	67.24	775	87.0

These tests were conducted keeping the oil temperature constant at 185 degrees to 190 degrees, and the pressure constant at 165 pounds, varying the steaming rate by varying the number of burners, and not by varying the quantity of oil per burner. The results were disappointing and the tests were discontinued after the third. The superheat was abnormally high, the stack temperatures were high, and the evaporation was very low. No improvement in the pressure drop was observed as a result of the alteration to the distribution pipes in the lower superheater drums. The weather conditions during the tests were very unfavorable. A pouring rain was falling during most of the time. The carbon deposit was substantially the same as in the previous test.

Before the next series of tests, the baffling was altered as shown on sketch, Plate 6. The plate baffles between the lower superheater drums and the generating tubes were replaced, except that a gas passage, about 2 inches wide was left adjacent to the drums. The lane baffles were arranged as during tests Nos. 38–6 to 38–8. The holes in the distributor pipes in the lower superheater drums were again enlarged, this time to 9/16 of an inch. In order to determine the cause of the

heavy carbon deposits, it was decided that, before conducting any further tests, a better grade of oil should be obtained. Accordingly a quantity of special oil was purchased. This oil, as the analysis shows, was particularly good. The following results were obtained:

Test No.	Duration, hrs.	No. of burners	Lbs. oil per sq. ft. H. S. per hr.
38–12	3	5	.503
38–13	2	9	1.005
38–14	3	9	1.385
38–15	2	4	.250
38–16	1½	7	.748
38–17	2	9	1.006

Test No.	Lbs. Oil per Cu. ft. F. V. per hr.	Eq. Evap.	Eff.	Stack Temp.	Superheat
38–12	4.740	15.456	76.84	647	47.3
38–13	9.483	14.633	72.61	813	78.6
38–14	13.051	13.876	69.03	874	83.4
38–15	2.354	15.439	76.79	535	24.4
38–16	7.053	15.308	76.20	712	54.7
38–17	9.487	14.372	74.64	740	66.1

These results were considered quite satisfactory as regards efficiency, superheat and stack temperatures, and it was decided that the arrangement of the baffling was the best that could be made. As a result of the better quality of oil, the furnace conditions were greatly improved, and the carbon deposit was very materially reduced. The carbon, however, was still quite bad, and it was decided that this was due to the boiler front, and that the condition could be corrected only by installing a different front, with some other register and atomizer. The weather conditions during the above tests were not very satis-

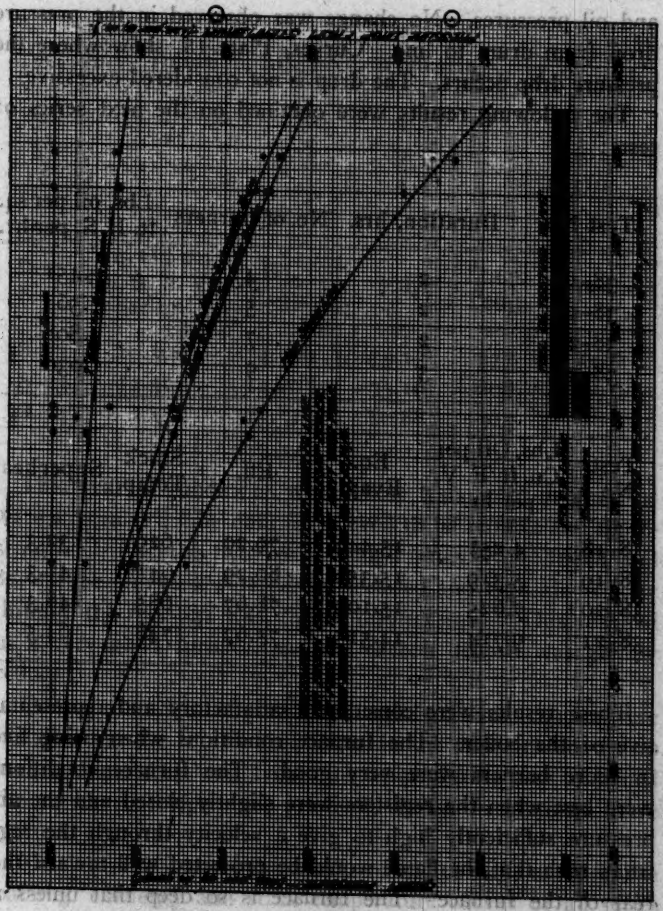


PLATE 13.

factory, and, therefore, it was decided to conduct additional ones for the purpose of obtaining more nearly representative data, and to observe the effect of various registers openings and oil pressures. No change was observed in the pressure drop from drum to line. Curves, Plate 13, show where the pressure drop occurs. The drop is not considered excessive.

The following results were obtained on the next series of tests:

Test No.	Duration, hrs.	No. of burners	Lbs. oil per sq. ft. H.S. per hr.
38–18	2	4	.451
38–19	2	5	.564
38–20	2	7	.808
38–21	3	9	1.030

Test No.	Lbs. Oil per Cu. ft. F. V. per hr.	Eq. Evap.	Eff.	Stack Temp.	Superheat
38–18	4.251	15.444	79.20	629	32.3
38–19	5.319	15.356	80.29	641	41.3
38–20	7.625	15.109	78.67	685	48.3
38–21	9.714	14.515	75.62	748	64.5

These results were considered satisfactory and representative of the boiler. The furnace conditions when using five or more burners were very good. The furnace conditions were best when the registers were slightly closed and the air pressure sufficiently high to give a velocity through the registers such that the flame would be projected well toward the rear of the furnace. The furnace is so deep that unless a high air velocity through the registers is used the distribution of gases in the rear of the furnace is bad.

To obtain points at very high and very low rates of combustion, one more series of tests were conducted with the following results:

Test No.	Duration, hrs.	No. of burners	Lbs. oil per sq. ft. H. S. per hr.
38–22	2	9	1.113
38–23	2	9	1.201
38–24	2	4	.370

Test No.	Lbs. Oil per Cu. ft. F. V. per hr.	Eq. Evap.	Eff.	Stack Temp.	Superheat
38–22	10.500	14.566	75.15	777	69.0
38–23	11.328	14.403	75.19	808	69.0
38–24	3.485	15.513	80.88	505	28.7

The weather conditions while these tests were in progress were favorable. The data is believed to be reliable and representative. For the two tests at high rates, Nos. 38–22 and 38–23, the registers were partially closed and high air velocity through the registers was, therefore, necessary. This resulted in a very good furnace, and the carbon deposit was greatly reduced.

During tests Nos. 38–19, 20, 23 and 24, thermo couples were installed (in the left nest of tubes, in the positions shown on sketch, Plate 14) and, from the temperatures thus obtained the curves of temperature gradient, shown on Plate 15, were plotted.

To show the progressive results of the changes made in the baffling, efficiency and stack temperature curves of all series of tests are plotted on one sheet, Plate 16.

The effect of weather conditions on the performance of the boiler was forcibly brought out. When it was raining, foggy

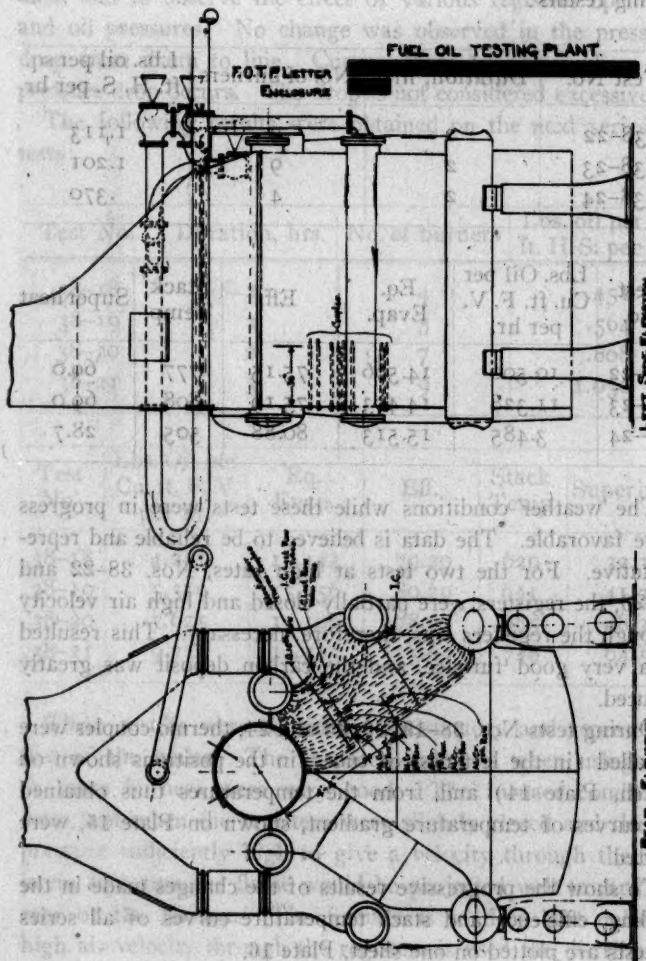

FUEL OIL TESTING PLANT.
F.O.T.P. Letter
Enclosure
Left Side Elevation.
Rear Elevation.

PLATE 14.

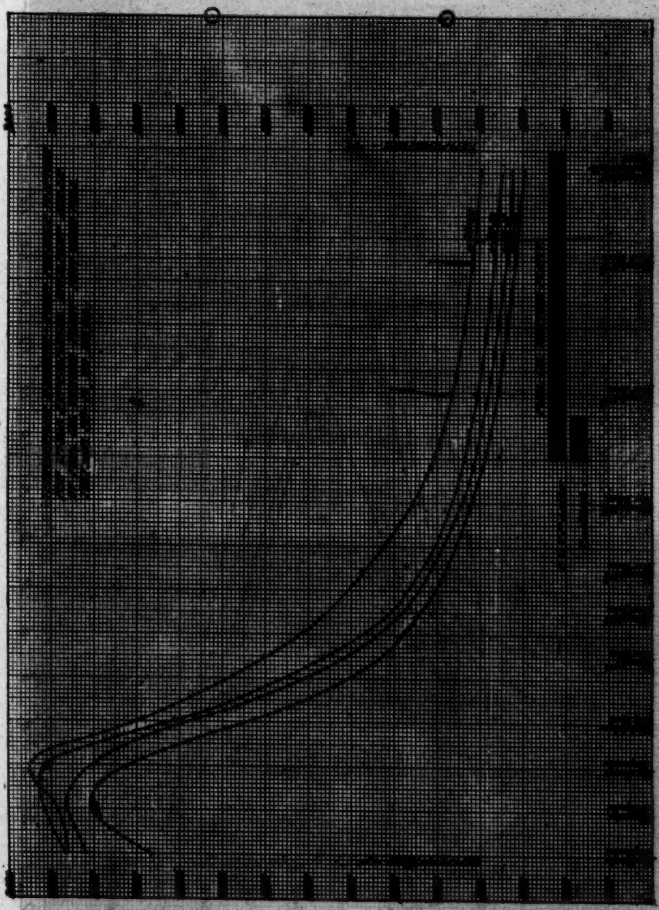


PLATE 15.

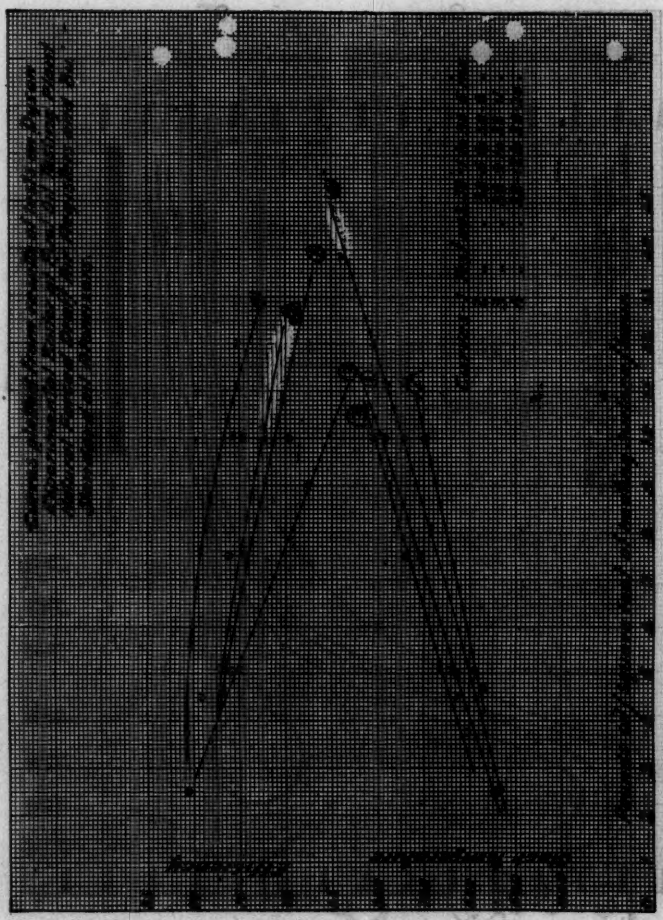


PLATE 16.

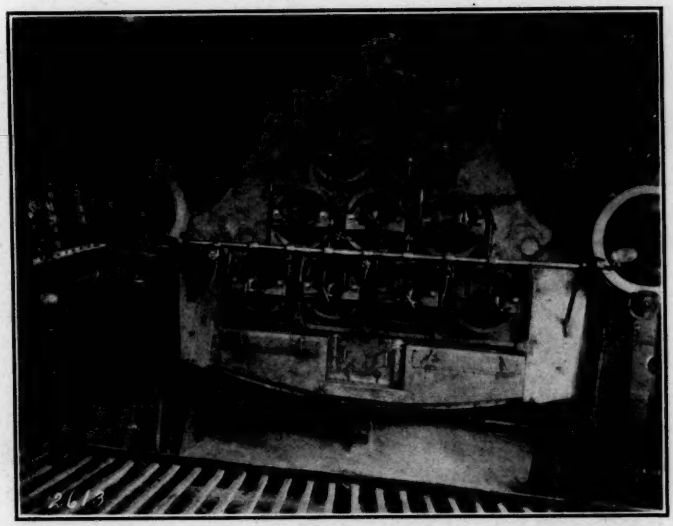


PLATE 17.

PLATE 18.

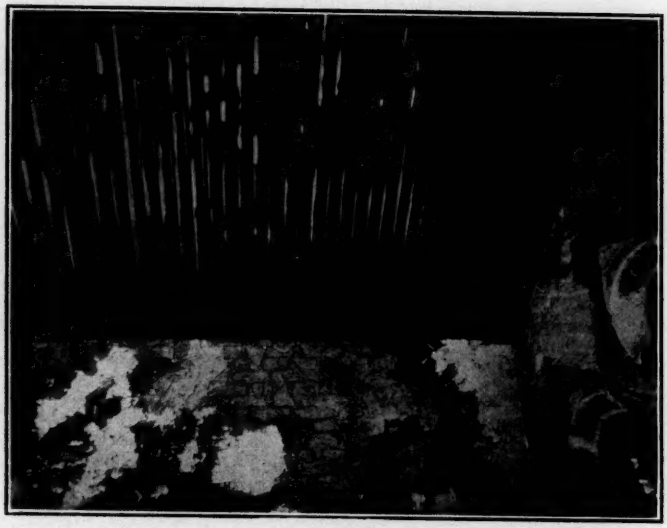


PLATE 19.

PLATE 20.

or misty, and, therefore, when the air contained much unevaporated moisture, the efficiency of the boiler fell off to a marked degree.

Plate 17 is a photograph of the boiler front as rigged during these tests. Plates 18, 19 and 20 are photographs which were made after these tests had been completed and the boiler had been equipped with five Lodi registers. They give a good idea of the appearance of the furnace, and show the condition of the tubes at the completion of the tests.

CONCLUSIONS.

These tests indicate that this boiler will prove a highly satisfactory steam generator for capital ships. It meets all the specified requirements for such a boiler, and shows a somewhat better evaporative efficiency than other boilers of its class which have been tested. It is believed that with a different boiler front, using a smaller number of larger registers and burners, improved operating conditions, especially as regards carbon deposits, will be obtained.

PRACTICAL ENGINEERING CHARTS.

By K. F. Smith, Lieut.-Comdr., U. S. Navy, Member.

[Editor's Note.—A similar article on this subject appeared in the "United States Naval Institute Proceedings" but as the subject has been presented here in a somewhat different manner it has been deemed advisable to print it.]

In the practice of engineering most of us find that at times we have computations to make involving more or less complicated formulae. When such computations are frequent, a chart for graphic solution becomes almost a necessity. Most of us can plot charts of the simple right line formulae, but find difficulty in charting the more complicated types involving two or more variables.

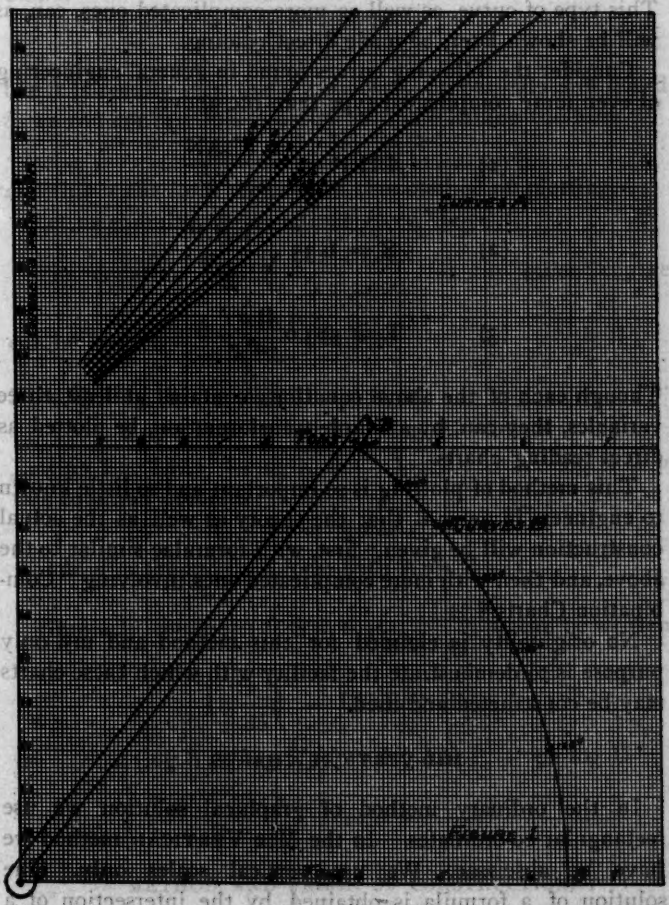
Unfortunately, most formulae used in engineering come far from being in the general mathematical shape indicative of a straight line.

$$y = ax + b$$

Thus, even for a simple chart, as for hourly coal consumption in tons, having given numbers and weights of buckets, we have a formula with two variables:

$$T = \frac{WN}{2240}$$

This is, of course, one of the simplest types and we can use the expedient of showing a series of straight lines for the ordinary weights of bucket, using as co-ordinates, Tons and Number of Buckets. This gives us a curve similar to A of Figure 1. We may use a swinging measuring arm, as in B—Figure 1, the auxiliary arm, OB, being placed on marks on the circumference, BC, corresponding to bucket weights and so located that $\theta = \cos^{-1} \frac{W_2}{W_1\sqrt{2}}$ where W_1 is the



weight of bucket for which the horizontal scale of the chart is constructed; W_2 the actual bucket weight.

Both of these are practical charts, but rather unwieldy. This type of curve, as well as more complicated ones, can, as will be shown later, be made direct reading.

Consider the following curves found in general engineering of even more complicated type than the above.

$$(1) \qquad \text{I.H.P.} = \frac{2\ \text{PLAN}}{33{,}000}$$

$$(2) \qquad S = 6.35 \sqrt{\frac{\text{I.H.P.}}{D^{3/8}}}$$

$$(3) \qquad T = 583.6 \frac{P\ a\ l}{d^2\ g}$$

Though each of the above equations contains at least three variables, they can, by a simple transformation, be plotted as direct reading charts.

This method of plotting is so important and so little known to engineers in general that the theory as well as the actual construction will be given; first, with formulae similar to the above, and then with more complicated ones involving "Combination Charts."

No originality is claimed for this method and my only purpose is to demonstrate the facility with which these charts may be constructed and used.

MID-VERTICAL CHARTS.

In the ordinary method of graphical solution we use rectangular co-ordinates. In the MID VERTICAL method we have no abscissae. We use vertical scales only. The solution of a formula is obtained by the intersection of a horizontal or inclined line drawn between certain points given in the data, with a vertical ordinate on which is laid off, to scale, the required unknown. By correlating two or

more such charts we may depict nearly any formula even with four or five variables and thus obtain a *Direct Reading Chart.*

The equation of a straight line is:

$$y = ax + b,$$

but it is readily seen that by choosing proper values of a and b, we can convert this equation into one having the form

$$Ax + By = C$$

and this is the form required in this method of plotting.

Let us examine the equations given above:

The equation $\text{I.H.P.} = \frac{2\ \text{PLAN}}{33{,}000}$ for any given engine can be written $\frac{\text{I.H.P.}}{K} = \text{P.N.}$

Now writing this in logarithmic form,

$$\text{Log I.H.P.} - \log K = \log P + \log N$$

Letting Log I.H.P. $- \log K = C$; $\log P = x$ and $\log N = y$, this equation becomes:

$$Ax + By = C$$

which is in the required form with $A = B = 1$.

Similarly, equation (2), $S = 6.35 \sqrt{\frac{\text{I.H.P.}}{D^{2/3}}}$, can be written,

$$\left(\frac{S}{6.35}\right)^2 = \text{I.H.P.} \times D^{3/2}$$

or $$2\ (\log S - \log 6.35) = \log \text{I.H.P.} + 3/2 \log D$$

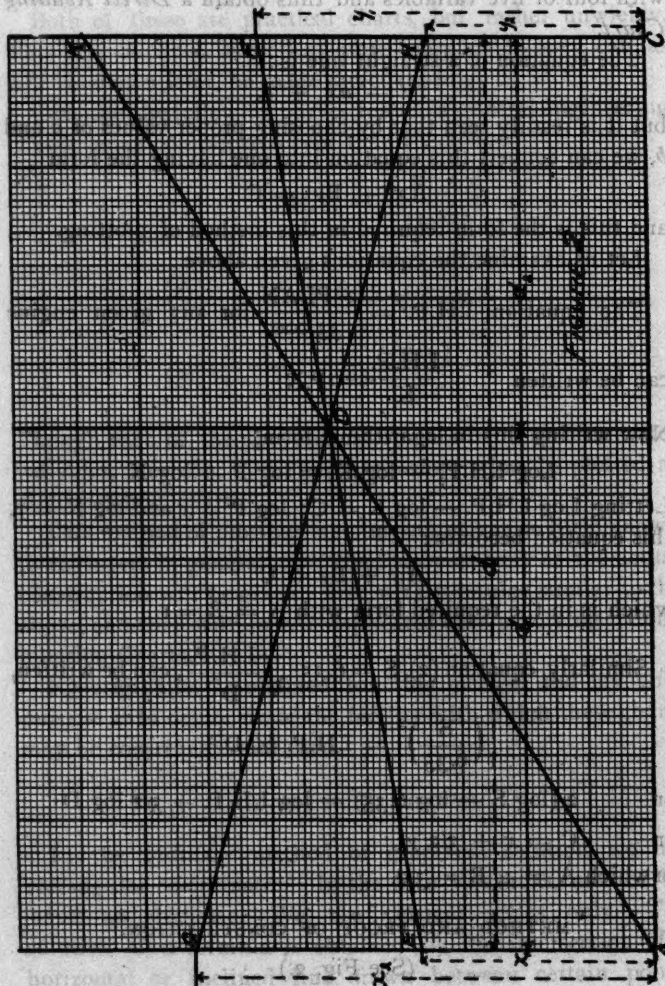
or $$C = x + 3/2\ y$$

in which $A = 1$, $B = 3/2$.

METHOD AND PROOF OF CONSTRUCTION.

(See Fig. 2.)

Erect two vertical lines AG and CK at any convenient distance $= d$ apart. Draw a horizontal line AC which will



be the zero line from which our measurements of x and y are taken. For simplicity, consider first A = B = 1 in the type curve and let AG be the X axis and CK be the Y axis.

In the equation $Ax + By = C$ assume a convenient value of C and compute two values x_1 and y_1 and x_2 and y_2 which will fulfill the equation.

Lay off AE (to the scale of L_1 units to the inch) equal to the value of x_1. Lay off CF (to the scale L_2 units to the inch) equal to the corresponding value of y_1. Connect these points by a straight line EF. Similarly lay off AG and CH, simultaneous values of x_2 and y_2 to the scale L_1 and L_2 respectively, and draw the straight line GH intersecting EF in D. Through D draw the vertical line BD.

Since AE, AG, CF, and CH are actual distances by construction,

$$AE \times L_1 = x_1$$
$$AG \times L_1 = x_2$$
$$CF \times L_2 = y_1$$
$$CH \times L_2 = y_2$$

Now, since C has been assumed,

$$x_1 + y_1 = C$$
$$x_2 + y_2 = C$$

(1) $(L_1 \times AE) + (L_2 \times CF) = C$

(2) $(L_1 \times AG) + (L_2 \times CH) = C$

(3) $AG = AE + EG$ (by construction)

(4) $CH = CF - HF$ (by construction)

Multiplying (3) by L_1 and (4) by L_2,

(5) $AG \times L_1 = AE \times L_1 + EG \times L_1$

(6) $CH \times L_2 = CF \times L_2 - HF \times L_2$

By construction $\frac{EG}{HF} = \frac{AB}{BC}$

(7) $HF = \frac{EG \times BC}{AB}$

Adding (5) and (6).

$$AG \times L_1 + CH \times L_2 = AE \times L_1 + EG \times L_1 + CF \times L_2 - \frac{EG \times BC \times L_2}{AB}$$

By substitution from (1) and (2),

$$C = C + EG\left(L_1 - \frac{BC}{AB} L_2\right)$$

To make this true,

$$EG\left(L_1 - \frac{BC}{AB} L_2\right) \text{ must be } = O$$

Hence $$EG \text{ or } \left(L_1 - \frac{BC}{AB} L_2\right) \text{ must } = O$$

By construction EG is not zero,

so (8) $$L_1 = \frac{BC}{AB} L_2$$

Let $AC = d$; $AB = d_1$; and $BC = d_2$.

Then $$L_1 = \frac{d_2}{d_1} L_2$$

$$L_1 + L_2 = \frac{d_2}{d_1} L_2 + L_2 = \frac{(d_2 + d_1)}{d_1} L_2$$

and, since by construction $d_1 + d_2 = d$,

$$L_1 + L_2 = \frac{d L_2}{d_1} \text{ and } \frac{d_1}{d} = \frac{L_2}{L_1 + L_2}$$

and, similarly, $$\frac{d_2}{d} = \frac{L_1}{L_1 + L_2}$$

In this construction we could assume any pair of values for x and y, but as long as C remains constant and the scales for x and y, are not changed, $\frac{d_1}{d_2} = \frac{L_2}{L_1}$. Therefore the MID VERTICAL remains in the same position and D is found to

be fixed. All lines representing a solution of $x + y = C$ with C constant will pass through D. DB, therefore, represents C to some scale L_3.

$$DB \times L_3 = C$$

Substituting in (1) and (2),

$$(L_1 \times AE) + (L_2 \times CF) = DB \times L_3$$
$$(L_1 \times AG) + (L_2 \times CH) = DB \times L_3$$

Now where $x = O$, $y = C$, and the point K represents C to the scale of L_3.

Therefore $CK \times L_2 = C$

But $BD \times L_3 = C$

$$CK \times L_2 = BD \times L_3$$

But $$\frac{CK}{BD} = \frac{AC}{AB}$$

Therefore $$CK \times L_2 = \frac{AB}{AC} \times CK \times L_3$$

$$= \frac{d_1}{d} CK \times L_3$$

$$\text{or } L_2 = \frac{d_1}{d} L_3$$

$$\text{but } \frac{d_1}{d} = \frac{L_2}{L_1 + L_2}$$

Therefore $$L_2 = \frac{L_2}{L_1 + L_2} \times L_3$$

Therefore $$L_3 = L_1 + L_2$$

Therefore *the scale along the middle vertical must be the sum of the outer scales.*

If the formula coefficients A and B are considered in the equation $Ax + By = C$, it will be found that the only effect will be the "opening out" of the x and y scales, and, since the C scale is correlated, the C scale as well, and we will find, using the same nomenclature (i. e., $x = L_1$ units to the inch, $y = L_2$ units to the inch, and L_3, the scale of C), that

$$L_3 = A L_1 + B L_2.$$

The position of the MID VERTICAL is such that

$$\frac{d_1}{d_2} = \frac{BL_2}{AL_1}$$

These last two are general and for plotting must be remembered:

(1) $L_3 = AL_1 + BL_2$

(2) $\frac{d_1}{d_2} = \frac{BL_2}{AL_1}$

A PRACTICAL EXAMPLE.

The ex-German Battleship *Ostfriesland* has cylinders 43 inches, 63 inches, 67 inches and 67 inches in diameter, the rod diameters are H.P. $7^3/_4$ inches, I.P. 8 inches, F.L. 8 inches, and A.L. $6^3/_4$ inches; the stroke is 48 inches.

Plot a horsepower curve for the three-engines combination from 5 to 15 knots, 30 to 83 R.P.M.

Mean effective pressure ranges from 5 pounds to 60 pounds.

$$\text{I.H.P.} = \frac{2\ \text{PLAN}}{33{,}000} = \text{KPN}.$$

where $$K = \frac{2\ LA}{33{,}000}$$

$$\log \text{I.H.P.} - \log K = \log P + \log N$$

$$\log \text{I.H. P.} - \log K = C$$

$$\log P = x \qquad Ax + By = C$$

$$\log N = y \qquad A = B = 1.$$

For convenience, I find it best to take millimeter cross section paper, or paper graduated 10 to the inch. We first decide on the size of the chart, having regard to the accuracy of the reading desired. In this case we decide on a chart 20 inches × 20 inches. We, therefore, draw our M.E.P. or x vertical at the left margin and our Revolution or y vertical at a distance of 20 inches to the right.

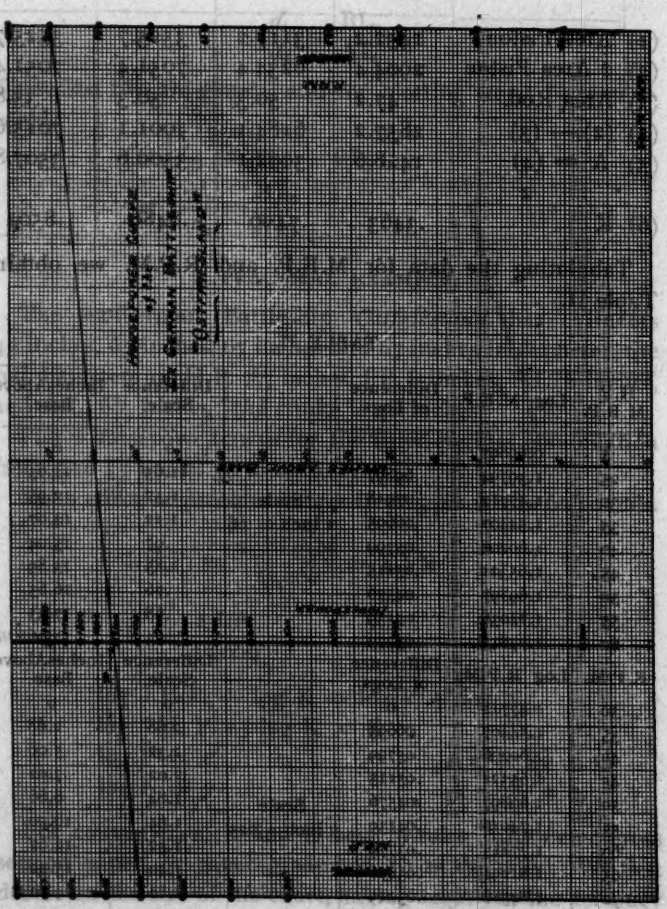
TABLE I.

		H.P.	I.P	F.L.	A.L.
(1)	Area Piston	1452.2	3117.2	3525.7	3525.7
(2)	2 Area Piston	2904.4	6234.4	7051.4	7051.4
(3)	Area Rod.	47.2	50.3	50.3	35.8
(4)	(2) — (3)	2857.2	6184.1	7001.1	7015.6
(5)	$A = \frac{(4)}{2}$	1428.6	3092.1	3500.6	3507.8
(6)	K	.3463	.7496	.8486	.8504

Tabulating the data for M.E.P. and R.P.M. we obtain Table II.

TABLE II.

H.P M.E.P.	Log M.E.P.	Difference of Logs		Difference Scale.	InchesAbove Base.
5	0.69897	0		0	0
25	1.39794	.69897		11.63	11.63
30	1.47712	.07918	Scale	1.32	12.95
35	1.54407	.06695	1 inch = .06	1.12	14.07
40	1.60206	.05799		.97	15.04
45	1.65321	.05115		.85	15.89
50	1.69897	.04576		.76	16.65
55	1.74036	.04139		.68	17.33
60	1.77815	.03779		.63	17.96
R.P.M.	Log R.P.M.	Difference of Logs		Difference Scale.	InchesAbove Base.
30	1.47712	0		0	0
35	1.54407	.06695		2.68	2.68
40	1.60206	.05799		2.32	5.00
45	1.65321	.05115		2.02	7.02
50	1.69897	.04576	Scale	1.84	8.86
55	1.74036	.04139	1 inch =.025	1.64	10.50
60	1.77815	.03779		1.52	12.02
65	1.81291	.03476		1.40	13.42
70	1.84510	.03219		1.28	14.70
75	1.87506	.02996		1.20	15.90
80	1.90309	.02803		1.12	17.02
85	1.92942	.02633		1.04	18.06



Examining the logs of M.E.P. and R.P.M. we see that the range of logs is 1.77815 — 0.69897 = 1.07918 for M.E.P. and 1.92942 — 1.47712 = .45230 for R.P.M.

We can now determine our x and y scales on the assumption that we have 20 inches of height available. If we choose 1 inch = .06 = L_1, we see 20 × .06 = 1.2 which is greater than the range of difference in the logs of M.E.P. Similarly 1 inch = .025 = L_2 and 20 × .025 = .5, which is greater than the range of difference in the logs of the R.P.M.

We are ready now to plot the H.P. chart.

We have determined L_1 and L_2 and, therefore, since the equation is in the form of

$$Ax + By = C \text{ and } A = B = 1,$$

$$L_3 = L_2 + L_1$$

$$L_3 = .025 + .06 = .085$$

We can now determine the position of the MID VERTICAL.

$$d = 20 = d_1 + d_2$$

$$\frac{d_1}{d_2} = \frac{.025}{.06}$$

$$\frac{d_2}{d_1} = \frac{.06}{.025}$$

$$20 = \frac{.025}{.06} d_2 + d_2$$

$$d_2 = \frac{1.2}{.085} = 14.12$$

and $$d_1 = 5.88$$

i. e., 5.88 inches to the right of X axis.

We now erect this MID VERTICAL 5.88 inches from the X axis.

We now lay off 5 on the base line at the bottom of the M.E.P. line and graduate the M.E.P. line by the distances above the base line shown in column six, (Table II), of the M.E.P.

Similarly we lay off 30 on the base line at the bottom of the R.P.M. line and lay off, on the R.P.M. line, the distances shown in column six (Table II) of the R. P. M.

CALIBRATION OF THE MID VERTICAL.

It now becomes necessary to graduate the MID VERTICAL. Our scale we have found to be 1 inch = .085, by computation, but we do not know where the zero lies.

Examining the formula,

$$\log \text{I.H.P.} - \log K = \log P \times \log N.$$

We assume any values we desire for P and N; say, for example, M.E.P. = 40 and R.P.M. = 80.

Then
$$\log \text{I.H.P.} - 9.53945 = 1.60206 + 1.90309$$
$$\log \text{I.H.P.} = 3.04460$$
$$\text{I.H.P.} = 1108.2 = \text{point P.}$$

If then we draw a line through R.P.M. = 80 and M.E.P. = 40 and where it cuts the MID VERTICAL (point P) plot 1108.2 I.H.P. we have the first point. Suppose, however, we take 1200 I.H.P. as our starting point for the I.H.P. curve.

$$\log 1200 = 3.07918$$
$$\log 1108.2 = \underline{3.04460}$$
$$\text{difference} \quad .03458$$
$$L_3 = .085$$

Therefore, we plot 1200 at a distance,

$$\frac{.03458}{.085} = .41 \text{ inches above point P.}$$

We now tabulate our I.H.P. curve as follows:

TABLE III.

I.H.P.	Log I.H.P.	Dif. of logs.	Difference Scale.	Inches above or below 1200 I.H.P.
1,500	3.17609	.02996	.35	+1.14
1,400	3.14613	.03219	.38	+ .79
1,300	3.11394	.03476	.41	+ .41
1,200	3.07918	.00000	.00	.00
1,100	3.04139	.03779	.45	—.45
1,000	3.00000	.04139	.48	—.93
900	2.95424	.04576	.54	—1.47
800	2.90309	.05115	.60	—2.07
700	2.84510	.05799	.68	—2.75
600	2.77815	.06695	.83	—3.58
500	2.69897	.07918	.93	—4.51
400	2.60206	.09691	1.04	—5.55
300	2.47712	.12494	1.47	—7.02
200	2.30103	.17609	2.00	—9.02
100	2.00000	.30103	2.36	—11.38

TO USE THE SCALE.

Take the M.E.P. (from the indicator card) and the revolutions per minute and connect the points indicating these factors by a straight line. Where this line cuts the MID VERTICAL will be found the instantaneous value of the horse-power.

It will be noted that the position of the MID VERTICAL is not affected by a change in the value of K and will thus be the same for all cylinders. It will, however, be necessary to graduate the M.E.P. and R.P.M. lines for each cylinder, taking into consideration the value of K for different cylinders.

In place of computing distances, they could be laid off directly by using the A or C scales of a slide rule.

COMBINATION CHARTS.

We have seen in the preceding, how the simpler formulae may be plotted for direct reading, but when more valuables are introduced we must construct two or more charts and link them together.

For example, consider the formula

$$xy^3 = K\frac{z}{w}$$

We can construct two charts (see figure 3)

$$(1)\quad xy^3 = C$$

$$(2)\quad K\frac{z}{w} = C$$

and connect them. Thus, from (1)

$$\text{Log } x - 3 \log y = \log C$$

Let L_1 = scale of x

and L_2 = scale of y

and $A = 1 \quad B = 3$

Therefore $L_3 = L_1 + 3L_2$

and $$\frac{d_1}{d_2} = \frac{3L_2}{L_1}$$

which give the scale and distances of the MID VERTICAL for equation (1) to any scale we desire (see figure 3).

We now assume values of x and y and find the corresponding value of C and mark the point where the line joining simultaneous values of x and y cuts the MID VERTICAL with its value m.

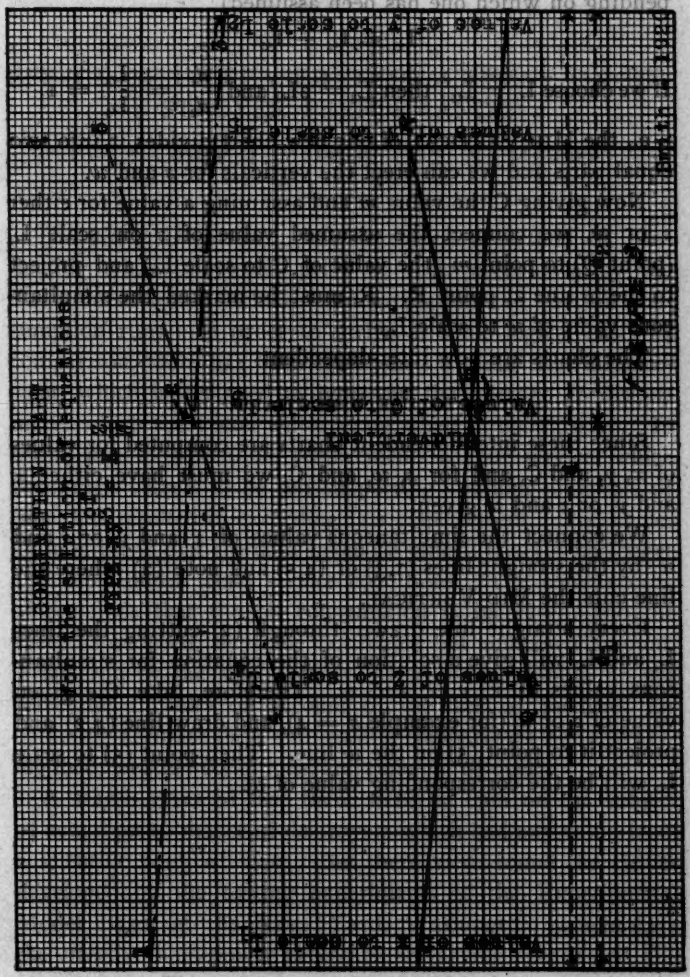
Now, examining the equation,

(2) $\log C - \log K = \log z - \log w$, and we again have the form

$$Ax + By = C.$$

We see $A = 1$; $B = -1$ (note the minus sign). If, however, we graduate the w vertically downwards instead of upwards, $A = B = 1$.

We next examine the range required for z and w and choose the scale of either L_4 or L_5 found most convenient. Maintaining the same scale for the MID VERTICAL of this



second equation, we compute the scale of either L_4 or L_5 (depending on which one has been assumed)

$$L_3 = L_4 + L_5$$

If we choose $L_4 = L_5$, then $L_3 = 2L_4$ and $\frac{d_1}{d_2} = \frac{L_4}{L_4} = 1$,

i. e., the MID VERTICAL of our new chart divides it into two equal parts and we can erect the verticals for z and w.

Now giving C the value m and assuming a value for either z or w, we connect the assumed value of z on scale L_4 (point Q) to point m (the value of C to scale L_3) and project to the w line at point R. R must be marked the simultaneous value of w to scale L_5.

The charts are now inter-dependent.

TO USE THE CHART.

Since these inter-dependent charts are computed for values of x, y, and C, and for z, w, and C, we must have either x and y, or z and w given.

We connect the simultaneous values of x and y, or z and w, by the straight lines (1,3) or (4,5) and note (2) where the line cuts the MID VERTICAL.

Every straight line drawn through (2) cutting the lines L_4 and L_5 will give a solution of the equation for simultaneous values of x, y, z, and w. Therefore, take the given value of z or w (for example $z = 4$), and draw line (4,2) and project it to point (5) on the w line. Then point (5) to scale L_5 will be the corresponding value of w.

EFFECT OF INCREASED DISPLACEMENT ON DESTROYER PERFORMANCE.

By Lt.-Comdr. J. McIrish, U. S. N., Member.

The Navy Department has developed a method of conducting trials of vessels prior to their acceptance by the Government which includes standardization and fuel oil consumption runs. In order to arrive at some logical basis of comparison, there has been established a mean trial displacement, which is designed to represent the average conditions under which the vessel will operate when in commission. In actual fact, however, the requirements of our service, which encourage vessels to be self-supporting for long intervals of time, has resulted in exceeding this displacement. The immediate effect is that the data taken on official trials is no longer applicable under usual service conditions. The commanding officer finds that the oil consumption is increased, and the speed is reduced, by some noticeable but unknown amount. Trials run by the ship's officers after commissioning do not solve the difficulty on account of the lack of facilities for obtaining accurate data and making comparisons due to varying load, weather conditions, and time out of dock.

A search revealed that there were no data on the subject of actual performance of destroyers which had any value from a scientific point of view. But there were found the curves developed by the towing experiments in the Experimental Model Basin at the Navy Yard, Washington, which gave the effective horsepower necessary at varying speeds and displacements. These data must be considered accurate and be used as the basis for further investigation, due to the scientific precision with which the data are obtained. This article

attempts to tie up the results obtained there with the results obtained on the trial course and has resulted in the development of the curves given in Plates I and II.

The new destroyers Nos. 75 to 347 can be divided into three groups depending on their under water body.

(A) The Bath Hull, which applies to the vessels built at Bath, New York Shipbuilding Corporation, The Wm. Cramps Sons' Ship and Engine Building Co., and the Navy Yard, Mare Island, California.

(B) The Fore River Hull, applying to boats built at Fore River, Squantum, and Union Iron Works.

(C) The Newport News Hull.

The tests at the Model Basin were run on each of these types, curves of speed *vs*, effective horsepower were developed for three different displacements. The method of using these data for finding the effect of change of displacement on shaft horsepower is the same for both types A and B and resulted in the curves of Plate I for the Bath Hull and Plate II for the Fore River Hull. On account of insufficient data and radical change in engine design of the Newport News vessels, no curves can be drawn at present for those boats. The method used is described in detail for the Bath Hull and tables also given for the Fore River Hull.

The most satisfactory method of obtaining the desired data would be to take a vessel of each class and conduct standardization and fuel consumption runs for all speeds at various displacements. The results could then be plotted, errors eliminated, and the effect of increased displacement would be shown immediately. This is too costly and lengthy to be possible, and it was necessary to resort to the results gotten on the trials of various vessels at different times. The effect of the varying conditions and lack of any attempt to locate the irregularities in the observed data resulted in many inconsistencies which became evident when direct comparisons were made. For example, comparison of the shaft horsepower of

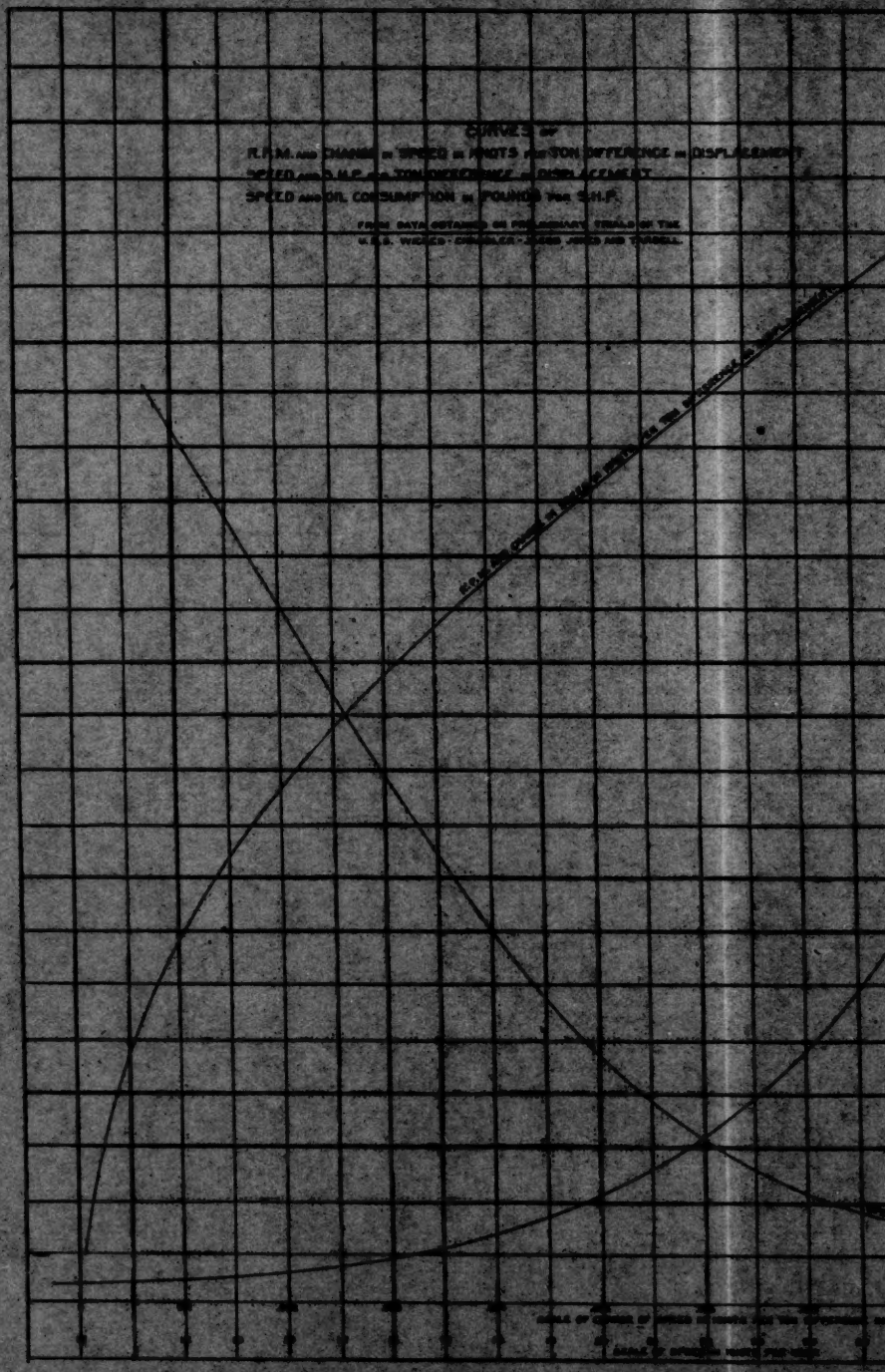

CURVES OF
R.P.M. and CHANGE in SPEED in KNOTS from TON DIFFERENCE in DISPLACEMENT
SPEED and H.P. for TON DIFFERENCE in DISPLACEMENT
SPEED and OIL CONSUMPTION in POUNDS per S.H.P.
FROM DATA OBTAINED ON PRELIMINARY TRIALS OF THE
U.S.S. WICKES - DESTROYER - 1918, AKRON AND TWOBELL.

PLATE I.

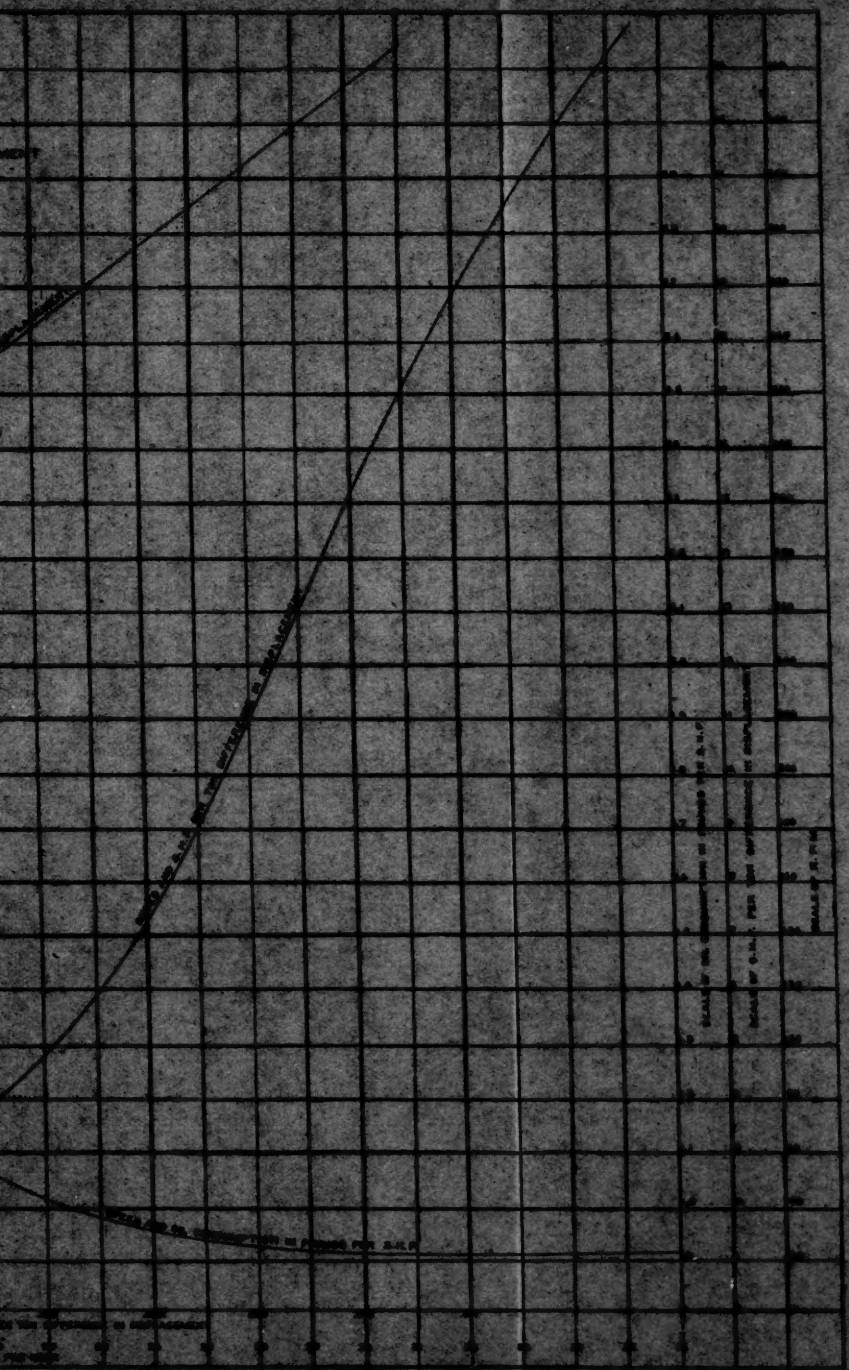


PLATE I.

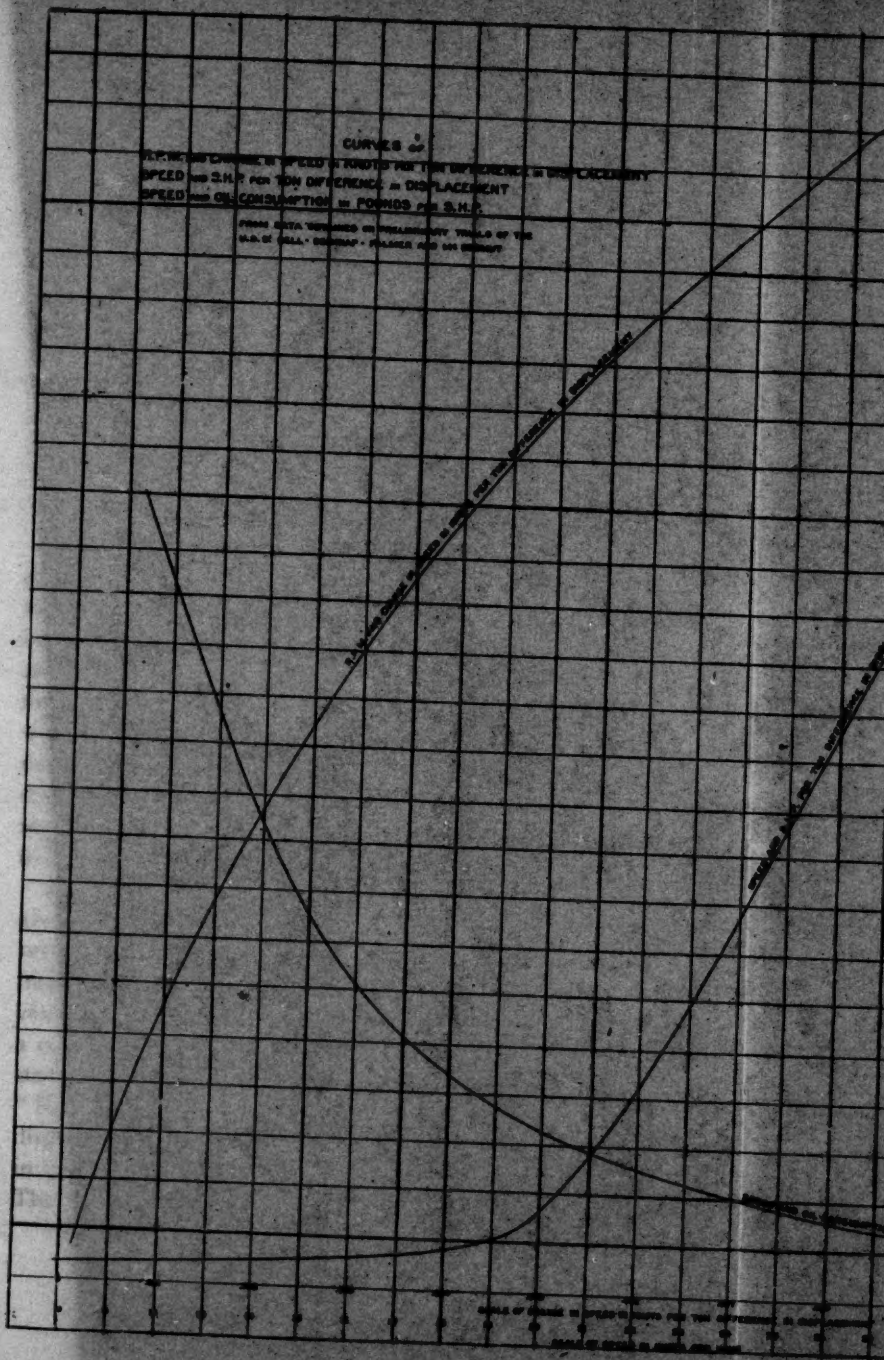
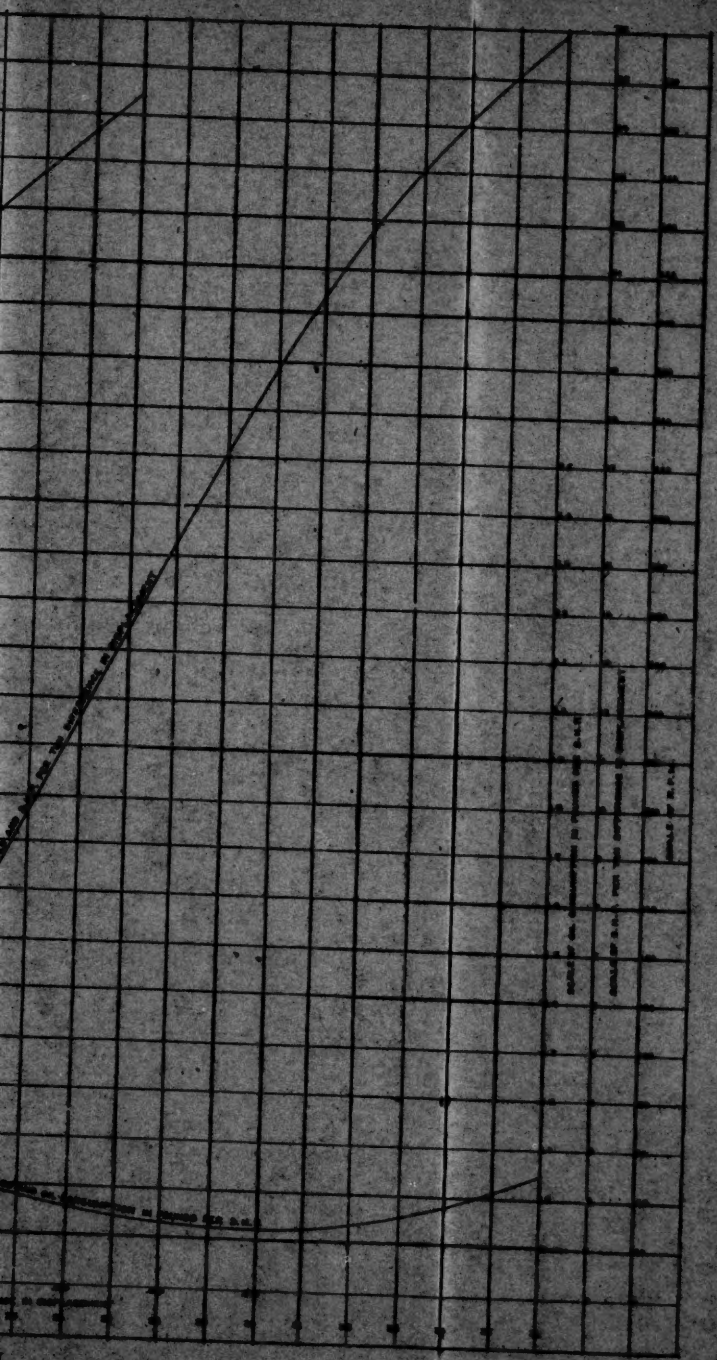

CURVES of
SPEED and S.H.P. FOR TON DIFFERENCE in DISPLACEMENT
SPEED and OIL CONSUMPTION in POUNDS PER S.H.P.

PLATE II.



the *Tarbell* and *Chandler*, which have the same under water hull but displacing respectively, 1,163 and 1,239 tons, give the following table:

TABLE No. I.

	S. H. P.		Diff. in S. H. P.	Diff. in S. H. P. per ton diff. disp.
	Tarbell	*Chandler*		
35	24,590	25,660	1,070	14.1
30	15,175	16,330	1,155	15.2
25	7,835	7,880	45	.59
20	3,210	3,140	(−) 70	(−) .92
15	1,240	1,225	(−) 15	(−) .197

A glance at the last column of this table, which is the difference in shaft horsepower per ton difference in displacement, immediately shows that the observed shaft horsepower in some instances are incorrect, and yet all the data were taken with the same care and worked out with the same methods. It is obviously impossible to say which reading is wrong and how much any one should be changed. Such variations occur in the results of all trials and are evidently due to instrumental errors, and the necessity arose of developing a method of comparing the observed data with data known to be accurate so as to indicate and eliminate the errors.

Due to the accuracy with which the standardization trials are run, comparison of the speeds obtained for the same revolutions per minute showed directly that the effect of increasing the displacement is to reduce the speed of the vessel by a definite amount. Therefore, for any given R.P.M. a correction must be made to the speed picked off from the standardization curve of the official trials. The curve marked "R.P.M. and change in speed in knots per ton difference in displacement" on Plate I is derived by comparing the speed in knots obtained for certain R.P.M. on the different vessels. The data are given in Table II, the speed being picked off

TABLE NO. II—R. P. M. VS SPEED.

R. P. M.	Knots				*Wickes* vs. *Chandler*		*Tarbell* vs. *Chandler*		Average
	Wickes 1,142	*Tarbell* 1,163	*Jacob Jones* 1,172	*Chandler* 1,239	Diff.	Diff. in speed per ton diff.	Diff.	Diff. in speed per ton diff.	
100	9.4	9.42	9.5	9.28	0.12	.00124	0.14	.00185	.00154
200	18.5	18.47	18.58	18.30	0.20	.00203	0.17	.00224	.00213
300	26.2	26.12	26.2	25.74	0.46	.00474	0.38	.00500	.00487
400	33.2	33.1	33.0	32.45	0.75	.00773	0.65	.00855	.00818
450	35.9	35.8	35.5	35.00	0.90	.00927	0.80	.01050	.00988

for 100, 200, 300, 400 and 450 R.P.M. and the difference in speed is divided by the difference in tons displacement for the various vessels. A factor is thereby obtained, based on the R.P.M. actually made, which when multiplied by the difference in displacement over the trial displacement will give the loss in speed in knots. This correction applied to the speed obtained for the R.P.M. on the official standardization will give the true speed of the vessel, and this true speed should be used in finding the factor from the curve marked "Speed and S.H.P. per ton difference in displacement" for the S.H.P. correction. A noticeable discrepancy occurs when the points for the *Jacob Jones vs. Chandler* are plotted; and in view of the fact that the weather and wind conditions were unfavorable at the time of the *Jacob Jones'* standardization, it is believed that these points may be disregarded when they fail to agree with the other vessels. The difference, though slight, is largest at the low speeds, but no great error is introduced in the final results by neglecting them.

For the Bath Hull, the following boats received full trials: *Wickes, Jacob Jones, Tarbell* and *Chandler,* and at speeds of approximately 35, 30, 25, 20 and 15 knots, from which curves were constructed so that the data could be accurately obtained for any desired speed. From the Model tank curves of E.H.P. there was picked off the E.H.P. for each vessel and each speed, given in Table III, Column 2; and for the same speed, from the data on official trials, the S.H.P. measured is given in Column 3; from Columns 2 and 3 was obtained the propulsive coefficient for each vessel at each speed which is given in Column 4. The propulsive coefficient for each speed and for all vessels are assembled in Table IV and averaged. The average propulsive coefficient obtained is entered in Column 5 of Table III. Using this average coefficient and the E.H.P. in Column 1, a new S.H.P. was obtained called the "computed shaft horsepower" and entered in Column 6. This computed S.H.P. is based on the E.H.P., and on account of using the average propulsive coefficient, due weight is given to the

TABLE No. III.

Wickes—1,142 Tons.

1	2	3	4	5	6	7	8
Knots	E. H. P.	S. H. P.	Pro. Coeff.	Ave. Coeff.	S. H. P. (Comp.)	S. H. P. Diff.	Per cent Diff.
35	13,860	23,950	.579	.575	24,100	+ 150	+ 0.63
30	9,230	14,500	.636	.624	14,820	+ 280	+ 1.93
25	4,770	7,300	.653	.644	7,420	+ 120	+ 1.62
20	1,795	2,700	.665	.633	2,925	+ 225	+ 8.33
15	695	920	.755	.676	1,030	+ 110	+ 11.01

Tarbell—1,163 Tons.

35	14,100	24,590	.573	.575	24,600	+ 10	+ .04
30	9,450	15,175	.623	.624	15,100	(−) 75	(−) .33
25	4,835	7,835	.618	.644	7,530	(−) 305	(−) 3.83
20	1,820	3,210	.568	.633	2,965	(−) 245	(−) 7.63
15	700	1,240	.564	.676	1,050	(−) 190	(−) 15.33

Jacob Jones—1,172 Tons.

35	14,200	25,150	.565	.575	24,800	(−) 350	(−) 1.39
30	9,520	15,300	.622	.624	15,250	(−) 50	(−) 0.33
25	4,880	7,500	.652	.644	7,580	+ 80	+ 0.93
20	1,830	2,950	.621	.633	2,970	+ 20	+ .67
15	705	920	.766	.676	1,052	+ 132	+ 14.38

Chandler—1,239 Tons.

35	14,940	25,660	.582	.575	26,350	+ 690	+ 2.69
30	10,060	16,330	.615	.624	16,280	(−) 50	(−) .31
25	5,085	7,880	.646	.644	7,980	+ 100	+ 1.28
20	1,900	3,140	.605	.633	3,115	(−) 25	(−) .80
15	730	1,225	.595	.676	1,090	(−) 135	(−) 11.01

TABLE NO. IV.

Propulsive Coefficients.

Knots	*Wickes*	*Jacob Jones*	*Tarbell*	*Chandler*	Average
35	.579	.565	.573	.582	.575
30	.636	.622	.623	.615	.624
25	.653	.652	.618	.646	.642
20	.665	.621	.568	.605	.615
15	.755	.766	.564	.595	.670

actual efficiency of the machinery and to the operating conditions. A comparison of Column 6 and Column 3 gives the variation of the computed shaft horsepower with observed shaft horsepower, and the difference is given in Column 7 while the percentage of error of this difference with the observed S.H.P. is given in Column 8. Columns 3, 6, 7 and 8 are reassembled for the different speeds in Table V so that a comparison can be readily made; and it will be noted that the error is largest for the 15- and 20-knot points, but in no case is larger than that which can be logically attributed to instrumental errors. In actual fact, the true cause is probably found in the method used in running the trials in which the vessel is ballasted so that the "mean trial displacement" is obtained when in the middle of the high speed runs. As the lowest speeds, beginning with the 15 knots, are first conducted, the displacement at this point is, therefore, somewhat greater than the "mean trial displacement," the resulting S.H.P. is consequently greater than would actually be obtained if the runs had been made at exact "mean trial displacement."

With the S.H.P. computed and compared, we are now in a position to make the necessary comparisons and find the true effect of the increased displacement. In order to obtain the widest range possible, the *Wickes* displacing 1,142 tons and the *Chandler* displacing 1,229 tons are used, giving a change in displacement of 97 tons. These are given below in Table VI, Column 3.

TABLE No. V.

	35 knots				30 knots			
	S. H. P. Measured	S. H. P. Computed	H. P. Diff.	Per cent Error	S. H. P. Measured	S. H. P. Computed	H. P. Diff.	Per cent Error
Wickes	23,950	24,100	+ 50	+0.63	14,500	14,820	+ 280	+1.93
Jacob Jones	25,150	24,800	(−) 350	−1.39	15,300	15,250	(−) 50	− .33
Tarbell	24,590	24,600	+ 10	+ .09	15,175	15,100	(−) 75	− .49
Chandler	25,660	26,350	+ 690	+2.69	16,330	16,280	(−) 50	− .31
	25 knots				20 knots			
	S. H. P. Measured	S. H. P. Computed	H. P. Diff.	Per cent Error	S. H. P. Measured	S. H. P. Computed	H. P. Diff.	Per cent Error
Wickes	7,300	7,420	+120	+1.62	2,700	2,925	+225	+8.33
Jacob Jones	7,500	7,580	+ 80	+ .93	2,950	2,970	+ 20	+ .67
Tarbell	7,835	7,530	−305	+3.83	3,210	2,965	−245	−7.63
Chandler	7,880	7,980	+100	+1.28	3,140	3,115	− 25	− .80
	15 knots							
	S. H. P. Measured	S. H. P. Computed	H. P. Diff.	Per cent Error				
Wickes	920	1,030	+110	+11.93				
Jacob Jones	920	1,050	+132	+14.38				
Tarbell	1,240	1,052	−190	−15.33				
Chandler	1,225	1,090	−135	−11.01				

TABLE No. VI.

Knots	1	2	3	4
	Wickes (1,142) S.H.P. (Comp.)	*Chandler* (1,239) S.H.P. (Comp.)	Diff. S. H. P. (97)	S. H. P. Diff. per ton diff.
35	24,100	26,350	2,250	23.2
30	14,820	16,280	1,460	15.05
25	7,420	7,980	560	5.78
20	2,925	3,115	190	1.96
15	1,030	1,090	60	0.62

The actual difference in S.H.P. shows an increase in shaft horsepower with the speed, such as could be logically expected, and this difference divided by the actual difference in displacement gives the data for the curve marked "Speed and S.H.P. per ton difference in displacement" on Plate I. Knowing the true speed, from this curve can be obtained a factor which is the S.H.P. difference per ton difference in displacement. Knowing the difference between the actual displacement and the trial displacement for that vessel, the increase in shaft horsepower is found by multiplying the ton difference in displacement by this factor.

The effect of change of displacement on oil consumption was analyzed on the basis that since the shaft horsepower increased with displacement, then the oil consumption should increase with the shaft horsepower developed. Using the computed shaft horsepower in order to eliminate instrumental errors and the total oil per hour at the various speeds, the curve marked "Speed and oil consumption in pounds per S.H.P." was plotted from the data given in Table VII.

From this curve, using the true speed, a factor can be found which is the pounds per hour per shaft horsepower. This factor multiplied by the total S.H.P. will give the fuel consumption in pounds per hour.

TABLE No. VII.

Knots	*Wickes*			*Tarbell*			*Jacob Jones*		
	Lbs. per hour	S. H. P. (Comp.)	Lbs. per hr. per S. H. P.	Lbs. per hour	S. H. P. (Comp.)	Lbs. per hr. per S. H. P.	Lbs. per hour	S. H. P. (Comp.)	Lbs. per hr. per S. H. P.
35	19,330	24,100	.8025	22,435	24,600	.913	24,850	24,800	1.000
30	13,050	14,820	.880	13,875	15,100	.917	14,430	15,250	.947
25	6,525	7,420	.879	7,913	7,530	1.05	7,950	7,580	1.048
20	3,660	2,925	1.250	4,090	2,965	1.378	4,060	2,970	1.365
15	1,965	1,030	1.905	2,100	1,050	2.000	2,145	1,052	2.038

	Chandler			Lbs. Per Hour Per S. H. P.				
				Wickes	*Tarbell*	*Jacob Jones*	*Chandler*	Average
35	24,412	26,350	.9275	.8025	.9130	1.000	.9275	.911
30	14,550	16,280	.894	.880	.917	.947	.894	.910
25	7,413	7,980	.928	.879	1.050	1.048	.928	.976
20	3,960	3,115	1.182	1.250	1.378	1.365	1.182	1.269
15	2,010	1,090	1.845	1.905	2.000	2.038	1.845	1.947

TABLE No. VIII.

Bell—1,182 Tons.

1	2	3	4	5	6	7	8
Knots	E. H. P.	S. H. P.	Pro. Coeff.	Ave. Coeff.	S. H. P. (Comp.)	S. H. P. Diff.	Per cent. Diff.
35	14,585	27,760	.525	.526	27,700	(−) 60	(−) 0.21
30	9,570	17,625	.543	.542	17,650	(+) 25	(+) 0.14
25	4,930	8,750	.563	.566	8,700	(−) 50	(−) 0.80
20	1,910	3,525	.542	.541	3,530	(+) 5	(+) 0.14
15	750	1,440	.521	.535	1,400	(−) 40	(−) 2.77

Palmer—1,190 Tons.

35	14,675	27,500	.533	.526	27,900	(+) 400	(+) 1.45
30	9,640	16,900	.570	.542	17,750	(+) 850	(+) 5.02
25	4,970	8,750	.569	.566	8,770	(+) 20	(+) 0.23
20	1,920	3,460	.555	.541	3,550	(+) 90	(+) 2.60
15	753	1,400	.537	.535	1,400	00	00

McDermut—1,225 Tons.

35	15,136	29,010	.522	.526	28,750	(−) 260	(−) 0.905
30	9,980	18,600	.526	.542	18,400	(−) 200	(−) 1.085
25	5,156	9,250	.557	.566	9,100	(−) 150	(−) 1.650
20	1,960	3,615	.543	.541	3,620	(+) 5	(+) 0.140
15	763	1,480	.515	.535	1,425	(−) 55	(−) 3.85

Belknap—1,254 Tons.

35	15,530	29,620	.525	.526	29,500	(−) 120	(−) 0.40
30	10,295	19,400	.530	.542	19,000	(−) 400	(−) 2.06
25	5,316	9,270	.573	.566	9,390	(+) 120	(+) 1.30
20	1,975	3,780	.523	.541	3,650	(−) 130	(−) 3.44
15	770	1,360	.565	.535	1,440	(+) 80	(+) 5.88

For any particular destroyer a more accurate determination of the fuel consumption can be arrived at by referring to the consumption curves for the standard vessel of her class. But as it was obviously impossible to include all these curves in this article, an average curve based on speed was determined as being more convenient and is given on the plates.

Having followed through the method of developing the curves for the Bath Hull, there are given below the similar Tables VIII to XII for the Fore River Hull, with the curves drawn from them, and similarly marked on Plate II.

TABLE No. IX.

Propulsive Coefficients.

Knots	*Bell*	*Palmer*	*McDermut*	*Belknap*	Average
35	·525	·533	·522	·525	·526
30	·543	·570	·526	·530	·542
25	·563	·569	·557	·573	·566
20	·542	·555	·543	·523	·541
15	·521	·537	·515	·565	·535

TABLE No. X.

Knots	S. H. P. (Comp.) *Bell* (1,182)	S. H. P. (Comp.) *Belknap* (1,254)	Diff. (72)	S. H. P. Diff. per ton diff.
35	27,700	29,500	1,800	25.0
30	17,650	19,000	1,350	18.75
25	8,700	9,390	690	9.59
20	3,530	3,650	120	1.67
15	1,400	1,440	40	0.555

To use the curves in practice, it is necessary to have the standard curves for the vessel of the class which was given full official trials, and to know at what displacement she was run, then knowing the R.P.M. which the engines are making and the actual displacement, all the remaining data can be found by following the procedure outlined below.

(*a*) Using the "R.P.M. and change in knots per ton difference in displacement," find the decrease in speed per ton increase in displacement.

TABLE No. XI.—R. P. M. vs. SPEED.

	Knots				*Bell-Belknap* (72)		*Palmer-Belknap* (64)		Average
R. P. M.	*Bell*	*Palmer*	*McDermut*	*Belknap*	Speed Diff.	Diff. in speed per ton diff.	Speed Diff.	Diff. in speed per ton diff.	Diff. in speed per ton diff.
100	8.97	8.95	8.7	8.9	.07	.00097	.05	.00078	.00088
200	17.72	17.75	17.58	17.55	.17	.00236	.20	.00313	.00325
300	25.15	25.05	24.92	24.92	.25	.00347	.15	.00235	.00372
400	31.78	31.85	31.55	31.30	.48	.00666	.55	.0086	.00600
450	34.70	34.70	34.48	34.20	.50	.00695	.50	.0078	.0074

	Bell-McDermut (43)	
100	0.13	.0030
200	0.14	.00326
300	0.23	.00535
400	0.23	.00535
450	0.22	.00510

TABLE No. XII (OIL CONSUMPTION).

Knots	*Bell*			*Palmer*		
	Lbs. per hour	S. H. P. (Comp.)	Lbs. per hour per S. H. P.	Lbs. per hour	S. H. P. (Comp.)	Lbs. per hour per S. H. P.
35	27,090	27,700	.978	28,200	27,900	1.010
30	16,740	17,650	.948	15,450	17,750	.870
25	8,688	8,700	.998	8,650	8,770	.985
20	4,040	3,530	1.145	3,920	3,550	1.102
15	2,325	1,400	1.660	2,160	1,490	1.450

Knots	*McDermut*			*Belknap*			Average
	Lbs. per hour	S.H.P.(Comp.)	Lbs. per hour per S. H. P.	Lbs. per hour	S.H.P.(Comp.)	Lbs. per hour per S. H. P.	Lbs. per hour per S. H. P.
35	28,200	28,750	.980	31,815	29,500	1.078	1.012
30	17,680	18,400	.960	17,565	19,000	.923	.925
25	9,170	9,100	1.007	8,812	9,390	.939	.982
20	4,130	3,620	1.142	4,100	3,650	1.122	1.128
15	2,370	1,425	1.662	2,235	1,440	1.550	1.556

(*b*) Multiply this by the difference in displacement, which will give loss in speed.

(*c*) Subtract this from speed gotten for the same R.P.M. on standardization; this will be the true speed.

(*d*) Using the true speed found in (*c*) and the curve marked "Speed and S.H.P. per ton difference in displacement," find the difference in S.H.P. per ton increase in displacement.

(*e*) Multiply this by the difference in displacement in tons, which will be the increase in S.H.P.

(*f*) Add this to the S.H.P. gotten for the same speed on the official trials. This will be the S.H.P. required.

(*g*) Using the true speed found in (*c*) and the curve marked "Speed and oil consumption in pounds per S.H.P.," find the pounds per hour per S.H.P.

(*h*) Multiply this by the total S.H.P. found in (*f*). This will be the oil required in pounds per hour.

To illustrate the above, the following examples are worked out:

"A"—For a vessel of the *Tarbell* class making 353 R.P.M. and displacing 1,400 tons, find the true speed, S.H.P. required, and the oil consumption.

For the *Tarbell:*

353 R.P.M. equals 30 knots;
S.H.P. for 353 R.P.M. is 15,180;
Mean trial displacement is 1,163 tons;
Pounds oil per hour (at 30 knots), 13,875.

(1) Using the curve marked "R.P.M. and change in speed in knots per ton difference in displacement" (Bath Hull, Plate I),
for 353 R.P.M., the speed change equals .0066;
the difference in displacement equals 1,400 — 1,163 = 237 tons;
.0066 × 237 = 1.564 knots;
30 — 1.564 = 28.44 knots, which is the true speed.

(2) Using the curve marked "Speed and S.H.P. per ton difference in displacement,"
for 28.44 knots the S.H.P. difference per ton difference = 11.77;
237 × 11.77 = 2,790 S.H.P. increase;
15,180 + 2,790 = 17,970 S.H.P. required.

(3) From the curve marked "Speed and oil consumption in pounds per S.H.P.," using 28.44 knots, we have pounds per hour per S.H.P. = 0.918;
17,970 × 0.918 = 16,496 pounds oil per hour.

"B"—For a vessel of the *Bell* class making 353 R.P.M. and displacing 237 tons more than on official trials, find the true speed, S.H.P. required, and the oil consumption.

For the *Bell:*
353 R.P.M. = 28.65 knots;
S.H.P. for 353 R.P.M. = 15,380;
Mean trial displacement = 1,182.

(1) Using Plate II,
for 353 R.P.M., the speed change = .00485;
.00485 × 237 = 1.16;
28.65 — 1.16 = 27.49 knots, which is the true speed.

(2) Using Plate II,
for 27.49 knots, the S.H.P. difference per ton difference = 14.25;
14.25 × 237 = 3,377;
15,380 + 3,377 = 18,757 S.H.P. required.

(3) Using Plate II,
for 27.49 knots, the pounds per hour per S.H.P. = 0.939;
0.939 × 18,757 = 17,610 pounds per hour.

The comparison in the two examples is interesting, showing in vessels of the two classes for the same increase in tonnage when making the same R.P.M. that the speed change is less

for the Fore River Hull, but the increase in its consumption is considerably higher on account of the greater horsepower required to drive the hull through the water.

By the above comparisons, we can state definitely that the effect of increasing the displacement is to decrease the speed and to increase the horsepower and the oil consumption. But no rough rule of thumb correction can be used in that the amount varies with speed and with the increase in displacement. The correction within close limits can be found from the curves.

In a demonstration of the practical usefulness of these curves, the writer undertook to predict the S.H.P. and oil consumption of the *Pruitt*, based on the data in this article.

The *Pruitt* was assumed to displace 1,210 tons, or 68 tons more than the *Wickes*. The predicted S.H.P. and oil consumption in pounds are compared below with those actually obtained on the *Pruitt's* official trials.

	S. H. P.		Oil, Lbs. Per Hour	
Knots	Predicted	Actual	Predicted	Actual
15	980	1,300	1,890	2,030
20	2,850	3,000	3,620	3,900
25	7,700	8,500	7,540	7,390
30	15,000	17,150	13,650	14,195
35	25,000	27,560	22,700	23,999

The predicted results agree so closely with the actual that a commanding officer could have gone to sea with no data except that provided by these curves and been entirely safe in his speed and oil consumption for any cruise.

In the same way a commanding officer leaving for a cruise on a higher displacement than that used on the trials of his vessel can predict his speed and fuel consumption for that draft, and if other conditions are equal, any difference can be laid to inefficient operation of the plant.

NOTES.

THE STILL ENGINE.

It is common knowledge, among those conversant with the internal combustion engine, that the greater part of the heat available from the fuel is carried away in the cooling water and the exhaust gases. Some loss also arises from radiation, while, if brake horsepower be taken as a basis for calculating the efficiency, friction must be brought into the account.

Very little scope exists for reducing the losses by radiation and friction, but more or less successful efforts to utilize the wasted heat in the exhaust gases were made many years ago, these taking the form of boilers in which steam was generated by passing the exhaust through them, while heated feed-water was derived from the cylinder jackets. In large plants some 2½ pounds of steam at 100 pounds pressure, per brake horsepower of the engines, has been generated in this way under favorable conditions. Assuming this steam to be used in an economical type of steam engine, the additional power obtained would be in the neighborhood of 15 per cent.

Up to that time the steam generated had been used either in separate cylinders or for heating purposes. In the Still engine it is employed to drive up the piston of the internal combustion engine, the cylinder being closed in at the bottom, while the piston is provided with a rod which passes through a stuffing box and gland as in an ordinary steam engine.

Many years of patient research work by Mr. W. J. Still, and those associated with him, more particularly Capt. Frank E. D. Acland, have resulted in the engine which forms the subject of this article. Mr. Still conceived the idea of going a stage further than using the jacket water to feed the exhaust boiler by making a boiler of the jacket. With the former arrangement only a small proportion of the water heated by the cylinder can be utilized; in the Still engine all the heat from the cylinder is usefully employed, excepting for a small percentage lost by radiation. At the same time the exhaust boiler is retained with the addition of a feed-water heater, the result being that by the time the exhaust gases are finally rejected their temperatures has been reduced from 900 to 150 degrees F. below which point no useful heat is practically recoverable.

ESSENTIAL FEATURES.

We will now describe the Still engine in detail with the aid of the sectional diagram shown in Fig. 1. Although this sketch, to some extent depicts the actual form of a Still engine, it is mainly diagrammatic. The Still engine may be of either the two or four-stroke type, and can run on gas, petrol or oil; in other words, the principle is applicable to all kinds of internal combustion engines.

Referring to Fig. 1, it will be seen that the upper part of the cylinder and the piston closely resemble these parts in other internal combustion engines. The engine shown is of the two-stroke type, in which, at the bottom of each stroke, scavenging air enters at one side of the cylinder, while the exhaust gases pass out at the other, through ports uncovered by the piston. The skirt of the latter is so long that only the upper part of the cylinder is exposed to the combustion of the fuel, the lower part being under steam.

On the right-hand side of the cylinder will be seen an ordinary vertical boiler, fitted with a liquid fuel burner. So far as the principles of the Still engine are concerned, this boiler might be fired with gas or coal. However, as a matter of convenience, it would be fired with gas or oil, according to the fuel used in the engine.

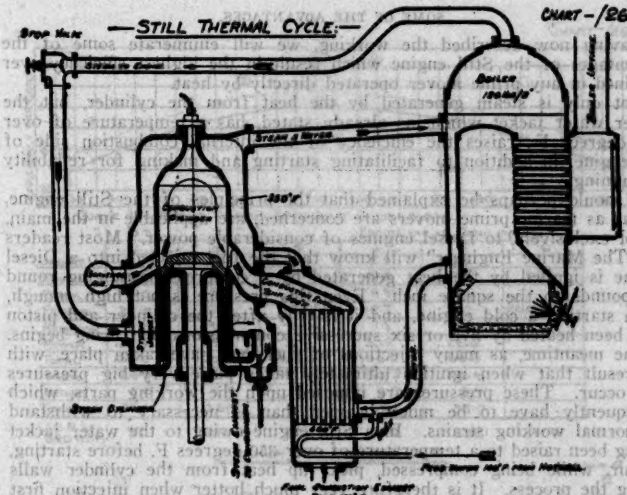


FIG. 1.

The Still engine starts on steam, raised in the above-mentioned boiler. Once the engine is running, fuel is shut off the boiler, in which steam is thenceforth maintained by the heat from the cylinder and exhaust gases. To make the principles of the Still engine clearer, a separate multitubular generator is shown in Fig. 1, through which the gases pass, and also through a feed-water heater below the boiler tubes, to be rejected finally at a temperature of about 150 degrees F. From the upper end of this exhaust generator, the steam produced circulates with the water into the jacket around the upper part of the cylinder, which is, of course, full of water. A steam pressure of from 120 to 150 pounds per square inch is maintained, corresponding with temperatures between 250 and 366 degrees F. The heat of the cylinder produces more steam, which passes off with that received from the exhaust generator to the vertical boiler. Thence it flows through the pipes shown, to the underside of the piston, by way of the steam chest surrounding the lower part of the cylinder. Naturally, valves have to be provided for admitting the steam to the cylinder and allowing it to exhaust at the correct times, as in an ordinary steam engine. On the combustion side of the piston, however, only a fuel-injection nozzle is required, the timing being controlled by the fuel pump, on the solid-injection system.

With the comparatively high temperature of the jacket water, the transfer of heat from the inside to the outside surface of the cylinder would be too slow if it were made of the usual thickness. To overcome this difficulty, the liner is made of one-third to a quarter of the ordinary thickness, but provided with numerous vertical ribs. These are turned on the outside and a steel barrel is shrunk over them, thus giving the necessary strength.

SOME OF THE ADVANTAGES.

Having now described the working, we will enumerate some of the advantages of the Still engine which result in the highest efficiency ever obtained in any prime mover operated directly by heat.

Not only is steam generated by the heat from the cylinder, but the hotter water jacket which, as already stated, has a temperature of over 350 degrees F., raises the efficiency of the internal combustion side of the engine, in addition to facilitating starting and making for reliability in running.

It should perhaps be explained that the principles of the Still engine, so far as marine prime movers are concerned, are applicable in the main, if not exclusively, to Diesel engines of considerable power. Most readers of "The Marine Engineer" will know that the fuel injected into a Diesel engine is ignited by the heat generated by a compression ranging round 450 pounds to the square inch. Even this pressure is not high enough, when starting a cold engine, and it is only after the cylinder and piston have been heated by five or six successive compressions that firing begins. In the meantime, as many injections of fuel may have taken place, with the result that when ignition ultimately takes place very big pressures may occur. These pressures are imposed upon the working parts, which consequently have to be much heavier than is necessary to withstand the normal working strains. In a Still engine owing to the water jacket having been raised to a temperature of over 350 degrees F. before starting, the air, while being compressed, picks up heat from the cylinder walls during the process. It is therefore very much hotter when injection first takes place, and the fuel is ignited with certainty on the first up-stroke. In fact, it has been found by experiment that ignition will take place with a compression of only 180 pounds per square inch, but 300 pounds has been adopted, as within limits the higher the compression pressure the greater the economy in fuel consumption per H.P.

It is interesting to note also that the operation of the steam side of the engine is more efficient than that in the usual type of steam engine. This is caused partly by the hot jacket, but mainly by the hot piston, the head of which is in contact with the burning fuel. The heat of the head travels down the skirt to some extent also. After the steam is cut off and allowed to expand, therefore, instead of partially condensing as in the ordinary engine, it takes up heat from the piston, and the steam develops more power than the theoretical maximum for saturated steam. Moreover, the process is beneficial to the internal combustion side by keeping the piston cool, so that larger diameters can be used without water cooling than for ordinary Diesel engines.

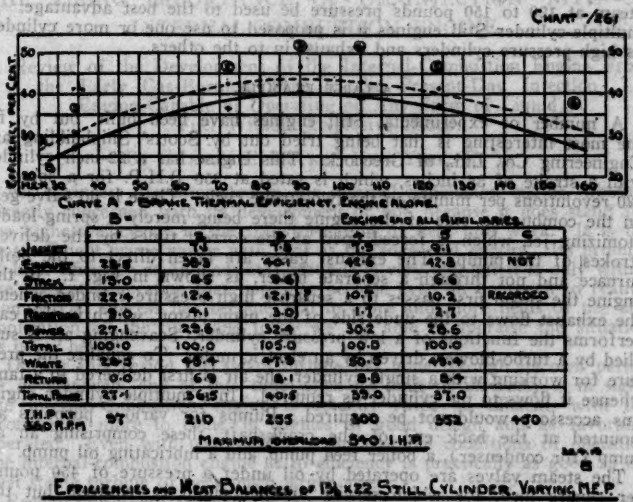
STARTING.

The Still engine is started under steam instead of compressed air, and maneuvers can be carried out for an indefinite period so long as some source of heat is applied to the boiler.

Some steam is used in almost all motor ships, and with the Still engine no separate boiler is required.

OVERLOADS.

A Still engine can be heavily overloaded if desired by starting the liquid fuel burner in the boiler and increasing the amount of steam available. For instance, an experimental engine of the two-stroke type, with opposed pistons, having a stroke of 22 inches in a 13-inch cylinder, was



	1	2	3	4	5	6
JACKET		10	7.8	7.5	9.1	
EXHAUST	28.8	38.3	40.1	42.6	42.8	NOT
STEAM	13.0	8.5	9.4	6.9	6.6	
FRICTION	22.4	12.4	12.1 I.P.	10.7	10.2	RECORDED
RADIATION	3.0	4.1	3.0	1.7	2.7	
POWER	27.1	29.6	32.4	30.2	28.6	
TOTAL	100.0	100.0	105.0	100.0	100.0	
WASTE	28.2	48.4	47.9	50.5	48.4	
RETURNS	0.0	6.9	8.1	8.8	8.4	
TOTAL LOSS	27.1	34.5	40.5	39.0	37.0	
I.H.P. AT 380 R.P.M.	97	210	235	300	352	420

MAXIMUM OVERLOAD 540 I.H.P.

EFFICIENCIES AND HEAT BALANCES OF 13 x 22 STILL CYLINDER, VARYING M.E.P.

FIG. 2.

made to develop 540 B.H.P. at 380 revolutions a minute, corresponding with mean effective pressures on the piston of 128.2 pounds for combustion and 57.9 pounds for steam, or a total of 186.6 pounds.

EFFICIENCY.

The engine quoted above has shown an efficiency of 40 per cent between the heat in the fuel and the brake horsepower (including all auxiliaries). So far as the writer is aware, so high an efficiency has never before been reached, even in a four-stroke engine. A figure often quoted as being the highest has been recorded by Mathot in connection with a 300 H.P. four-stroke engine, this being 36 per cent. It is calculated that, had this engine been converted into the Still type, the brake efficiency would have been no less than 44.4 per cent.

It should perhaps be stated that the highest efficiencies are not obtained with overloads, the 40 per cent quoted above having been realized with a combined M.E.P. of 90 pounds per square inch. This feature is well brought out in Fig. 2, showing the efficiency curve, and other particulars of this engine.

CONDENSING AND COMPOUNDING ON THE STEAM SIDE.

To obtain the greatest amount of work out of the steam, it is necessary to condense it, thus adding some 12 pounds or more per square inch to the M.E.P. by reducing the back pressure below the piston on the down strokes (due to the atmosphere) by this amount. Condensing the steam also greatly reduces its temperature, and thereby helps to cool the piston.

Only by compounding or adopting the triple expansion principle can steam at 120 to 150 pounds pressure be used to the best advantage. In multiple-cylinder Still engines it is proposed to use one or more cylinders as high-pressure cylinders and exhaust in to the others.

A LARGE EXAMPLE.

A number of experimental Still engines have been built, but by far the most interesting is that being tried out by Scotts' Shipbuilding and Engineering Co., Ltd., of Greenock. This engine has a 22-inch cylinder, with a stroke of 36 inches, which is rated at 350 B.H.P. for a speed of 120 revolutions per minute on the two-stroke cycle. There is no valve gear on the combustion side of the engine there being merely a spring-loaded atomizing jet, which is forced open at the correct times by the delivery strokes of the pump. The exhaust gases are taken direct to the boiler furnace and not through a separate boiler, as shown in Fig. 1. In this engine the steam first passes to a separate high pressure cylinder, whence the exhaust flows to the underside of the main piston, which in this case performs the functions of a low pressure cylinder. Scavenging air is supplied by a turbo-blower, driven by an electric motor. To equalize the pressure for working with a single cylinder, the air is first delivered to a tank, whence it flows to the cylinder as required. In a multiple cylinder engine this accessory would not be required. Pumps for various purposes are mounted at the back end of the crankshaft, these comprising an air pump (for condenser), a boiler feed pump, and a lubricating oil pump.

The steam valves are operated by oil under a pressure of 450 pounds per square inch, for which service another pump is required, but this arrangement is simply one method working the valves. Eccentrics, with link motion, might be used as in the ordinary marine engine.

Reversing gear is fitted, and both the steam and internal combustion sides of the engine are maneuvered by one hand-wheel. This engine has been adopted by Scott's as a standard unit, and will be built with any number of cylinders from three to eight. It is interesting to note that the cylinder wall is only 5/8 inch thick, against at least 2 inches in the ordinary Diesel engine.

For testing purposes, both a Froude brake and a dynamo are coupled to the crankshaft. Unfortunately, test figures are not yet available, but a consumption of .36 pound per B.H.P. hour, including auxiliaries, has been realized, with a gain in power of about 25 per cent.

GENERAL CONCLUSIONS.

No doubt many will contend that the accessories for the steam side of the engine make the Still engine a very complicated prime mover. But it must be remembered that neither air compressors nor valve gear for the combustion side are required. Moreover, the advocates of this engine claim that in spite of the steam accessories, the engine is lighter and takes up no more space than the ordinary Diesel type of similar powers and speeds, this being due to the higher powers realized for the cylinder dimensions, and the comparative lightness of the parts, owing to the elimination of excessive pressures at starting.

From whatever angle the engine is considered, the gain of 20 to 25 per cent in power on a given oil consumption, together with the big overloads available, would seem to far more than justify the extra gear. An invaluable feature also is the engine's ability to turn slowly on steam for any length of time.—"The Marine Engineer and Naval Architect," September, 1921.

THE RISING IMPORTANCE OF OIL-INJECTION TYPE OF INTERNAL-COMBUSTION ENGINE.

A Review of the Development of the Internal-Combustion Engine From the Early Gas-Burning Type to the Present-Day Injection Engine Capable of Operating on Any Form of Liquid Fuel Without Explosive Shock.

By Charles E. Lucke,* New York, N. Y.

All engineers are interested in the internal-combustion engine, whether directly concerned with its development or not, because it does stand for the highest efficiency in the transformation of heat into work. It is the most efficient prime mover with heat as the source of energy. Its promise of high efficiency on fundamental grounds is very old, but it is only within the present generation that hope has become even an approach to reality. Commercial success has become real in more and more fields of use and application all the time, and today we are able to appraise the situation as never before.

During this period the steam turbine has gone through its own period of development, and at this time when its limit of efficiency is in sight, the internal-combustion engine is actually twice as efficient, with the promise of more to come, in engines of good design and as favorably operated. Even with poor design or unfavorable conditions of operation the internal-combustion engine as to efficiency is practically as good as the best steam turbine of very much larger size.

Recognizing these facts, it is not difficult to understand the real and continuing interest in this problem of commercializing the internal-combustion engine. At the same time it is not at all clear just why it has developed along certain lines and not along others. Development along gas-engine lines—engines burning gas—has been a disappointment, and taking the world as a whole, the gas-burning engine has not been much of a financial success.

In spite of such a discouragement, however, the internal-combustion engine has gone ahead and is today becoming a dominant, if not the dominant, new factor in transportation. Where it has failed in stationary practice with gas as a fuel, the reason is to be found in its high first cost and maintenance charges as against the low first cost and maintenance charges of the steam-turbine plant, which factors do not equalize through fuel saving unless the fuel cost is very high per unit and the load factor also high, conditions which have not generally obtained. On the other hand, in the transportation field the liquid-fuel applications are the ones that have come into favor because of their peculiar adaptability and the impossibility of competition by the fixed steam and hydraulic central stations

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on land. Even on the sea we find the motorship making very substantial headway against the steamer equipped with our most modern turbines.

RISE OF IMPORTANCE OF LIQUID-FUEL DEVELOPMENT IN THE COMBUSTION-ENGINE FIELD.

This phase of the problem—the rise of importance of the liquid-fuel development in the combustion-engine field—is worth while analyzing. It is important for all engineers to know something about the business and problems of others, and on this assumption there should be general interest in the story of the adaptation of the internal-combustion engine to the uses of liquid fuel, so as to aid in the adaptation of the engines thus developed to transportation by the motorcycle, the automobile, the motor truck, the tractor, the railroad car or locomotive on land, by aircraft, and on the water by the motor boat and the motorship—in addition to certain stationary uses.

The first successful commercial machines of the internal-combustion class were gas-burning engines. While they have now sunk into a condition of more or less commercial insignificance, they have left behind a useful influence in that they have taught certain lessons that are of value in solving the problems of liquid-fuel adaptation.

The first lesson taught by the gas-burning engine is, the higher the efficiency the higher must the compression be. That is as it should be on thermodynamic grounds, and experience has amply demonstrated the validity of the theory. The second lesson is, that in addition to high compression, high efficiency is obtainable only if combustion is carried out in a correct and proper manner as to timing and rate. The gas-burning engine has also demonstrated that to get the maximum results in both power and efficiency it is equally necessary that the fuel be intimately and homogeneously mixed with the air throughout its entire mass, and that the cylinder be fully charged with that kind of mixture.

In addition to these principles of combustion for transforming high percentages of heat into work, the building of gas-burning engines has established many basic principles in the structural problem. To make cylinders, pistons and heads that will not crack is not as easy as it would seem, but taking the experience of the world at large, it can be said that reliable means of avoiding cracks have been devised.

In the adaptation of liquid fuel there are certain special problems that have to be faced that did not exist with the gaseous-fuel internal-combustion engine. The two principal classes of problems are in the fuel itself and in the special type of service to be met. It very early appeared there could be no such thing as a universal liquid-fuel engine equally good for gasoline, kerosene or fuel oil, or equally good for boats or automobiles or aircraft. There might very well be an automobile engine or a motorship engine, or a tractor engine, or a railroad engine, or a stationary electric-lighting-set engine, but each must be different.

It is this adaptation that occupies most of the period of development. To study the fuel phase of liquid-fuel adaptation, fuels must be divided into the two classes that are now found commercially but which division originally was not so clear. The first class includes fuels that are sufficiently volatile to make a more or less homogeneous and gaseous mixture with air by passing through so simple a device as a carburetor, which is similar to the older air-gas mixing valve of all gas-burning engines. The second may be termed the non-volatile class, and it includes anything that cannot be used in such a carburetor with or without heat, but which requires a device that must be built into part of the engine structure rather than a device or attachment to what would otherwise be a gas engine, thus initiating the injection oil engine.

The really difficult problem of the gasoline engine appears only when it is realized that the fuel available is no longer volatile enough to make the desirable homogeneous mixture, but not yet bad enough to require an injection engine. Before getting down to the problem of the injection engine proper, however, it is desirable to analyze some of the difficulties encountered in adapting the gas-burning engine and its principles of good utilization to light, and then to heavier, gasoline.

DIFFICULTIES ENCOUNTERED IN ADAPTING THE GAS-BURNING ENGINE TO LIGHT AND THEN TO HEAVIER GASOLINES.

The first principle of maximum compression cannot be carried as far as is desirable because the ignition temperature of these gasoline mixtures is lower than that of the gases forming the bulk of the fuels for the more efficient gas-burning engine. Furthermore, the temperature of the mixture before compression is no longer under the complete control it used to be with cold gas, and the temperature at the point of ignition or when compression ends is as much a function of the temperature before compression begins as it is of the amount of pressure rise.

The conclusions reached in gas-burning engine practice with regard to mixture quality, proportionality, homogeneity, intimacy, are all verified with gasoline. In proportion as that kind of mixture is attained with the volatile liquid fuel, so is it possible to attain some fair measure of the promise of efficiency, but the limit of attainment and the realization of it both fall off as the volatility falls off, and with the gasoline we are now using approximately only one-half vaporizes in the intake passages.

The other half that will not vaporize is carried along in three different states: (*a*) as a film on the walls, such as rain will form on a window pane (*b*) as a fog that floats, and (*c*) as a rain that is falling or driven by the air currents. A fog turns into rain, the rain makes films and the action cannot go far before the fuel is all wall film. No amount of ingenuity with devices such as screens, baffles and paddles can prevail, because these cannot overcome the laws of liquid flow and vapor pressure that are operating. This is a problem of serious importance, because if the unvaporized liquid fuel gets into the cylinder and strikes a hot spot, such as the piston head, it will vaporize there locally, and will form on the piston head a "pancake" of vapor that will have displaced the air and not be mixed with it. This when combustion takes place above it, is simply heated to decomposition temperature, producing carbon, and though actually not burning itself, it fouls up the engine and interferes with its operation with a loss of fuel.

If the unvaporized fuel entering the cylinder strikes on a cold cylinder wall, it will run down past the piston into the lubricating oil. This unvaporized fuel is mainly kerosene and the lubricating oil is also a petroleum product. They are mutually soluble and as a consequence the viscosity disappears. The mixture is no longer a lubricant. It runs down into the crankcase and destroys the lubrication of the main bearings and the crankpins, as well as the piston pins. How can these things be prevented from happening? Unless they are prevented, the engine is no longer commercial.

Two courses of action are open. The first is to heat the mixture as it leaves the carburetor, and thereby raise the vapor pressure to a point where in a 15 to 1 proportion there will be a vaporized-fuel and air mixture at the minimum possible temperature and a pressure of one atmosphere. It may be said, therefore, that a moderate amount of heating is permissible, and possibly a sufficient amount to completely dry the mixture if the gasoline is not too heavy, but complete heating for a kerosene mixture is not. Gain will be realized, but also a loss, and the loss will overbalance the gain, and the practice must be abandoned.

The next mode of attack is to try to handle the mixture with some of the fuel as a liquid. To handle a wet mixture means really, in the modern multi-cylinder engine, to distribute the stream of liquid as it runs along on the inside of the pipes, to four, six, or more branches, giving to each the same amount of liquid in order that all cylinders may work the same, assuming the liquid will be vaporized or sprayed as it enters each cylinder. To accomplish this it is necessary to know how the liquid moves. Imagine the liquid coming up the side walls of the riser and approaching a bend

FIG. 1. GASOLINE-KEROSENE CARBURETOR ENGINE, LIQUID ON MANIFOLD WALLS

as in Fig. 1. How would it turn? Observations in glass show that the liquid forms a very substantial lump at A, just beyond the turn, and on the inside of the bend. Films collect at the outside of the bend, but the velocity of the air-vapor mixture is so great at that point as to drive the liquid film around the bend to the point of least velocity. It is really the shape of the stream of air that the plug of liquid reveals.

Therefore, if the gasoline is not very heavy, then a moderate heating of the mixture—not too much—with some form of hot spot is the right thing so as to avoid preignition, and with this moderate heating a manifold to take care of the distribution of the rest of the liquid.

REMEDIES PROPOSED.

The net result of all this is that the gasoline carburetor engine is approaching a crisis in its history that is going to force the use of radical remedies. The remedies now being considered are as follows:

First, the elimination of the manifold entirely. This will take away the distribution problem and will permit the delivery of the liquid as a liquid with its air into the cylinder directly. No particular nozzle spraying is needed and not much vaporizing, because by properly forming the inlet valve and its passages a combustible foglike mixture will be formed as the charge enters the cylinder. This method has proved to be successful, and it is now the standard in use for all farm and most tractor engines, several hundred thousand of which are made every year, burning kerosene without any mixture heating whatever beyond what is incidental to suction.

Lubricating-oil contamination still is troublesome, and all such engines suffer from it to a greater or less extent. To minimize this the cylinder lubrication must be kept separate from that of the main bearings and crankpins. With admission of the mixture, however poorly vaporized, directly to the cylinder, good mixtures can be made without reduction of compression, but it is actually increased by adding water. This allows the water to enter the cylinder as a spray, just as the kerosene does,

and by reason of its thermal, and to some extent its chemical, action, the compression can be raised so that some tractor engines and many farm engines have as high as 90 pounds compression.

The second remedy is to change the volatility of the gasoline by mixture with another and more volatile fuel. By properly selecting the things to be added to the gasoline, it is possible to not only improve the volatility, but at the same time raise the ignition point. Experiments with a benzol-alcohol-kerosene mixture at Columbia University have yielded truly wonderful results judged by the possibilities of the future. It is amazing to what extent compression can be raised on a charge of gasoline ready to preignite normally, with a minor amount of alcohol added. This is also true with benzol. Such mixtures carried the German aircraft through the war.

ABANDONMENT OF THE CARBURETOR ENGINE AS A MEANS OF SOLVING THE PRESENT MOTOR-FUEL PROBLEM.

The third remedy—and this is the radical thing—is to abandon the mixture engine entirely and take up the injection engine. The abandonment of the carburetor engine, or the complete premixture engine, is a thing that would solve the present fuel problem as we hear of it, and that fuel problem is a real problem. America is today facing a situation in its liquid-fuel supply that is one of the most serious things that has happened industrially. This country has reached the point where imports exceed exports. That means a real shortage. It is due to the motor-car and allied demand. Its direct effect is increased prices for those volatile constituents that are in greatest demand, but without a corresponding increase in price for such residuals as are not in demand. A switchover from the carburetor engine, which requires the light distillates, to an injection engine—independent of the distillate and of a kind which is independent of volatility, operating with anything having proper fluidity—would mean that the automobile industry would be revolutionized. Such a step cannot be taken suddenly. It is a difficult job, but it is possible to explain the difficulties and to state the progress that is being made in solving them. Not only is this a matter of interest to the engineer, but it is a matter of national importance.

In case of the successful development of a suitable injection engine the problem of gasoline shortage disappears, because the injection engine can handle any petroleum distillate, any coal-tar product, any alcohol or similar fuel, subject to the one condition that it shall be of proper fluidity to pass the pump valves and spray finely at the spray orifice. If a fuel is not naturally of high fluidity, there is not one that cannot be made of proper fluidity by adequate heating. The temperature of heating, however, must not be carried so far as to cause a decomposition with carbonization and cracking.

INJECTION AND CARBURETOR ENGINES COMPARED.

The injection engine, compared with the carburetor engine, not only makes engines independent of the grade of fuel (there is only one requirement besides its viscosity—cleanliness), but it presupposes that there will be under compression only air, and that only after compression shall the fuel be injected. This means that by properly choosing the time of injection the compression may be as high as pleases the designer. There is no longer any limit of temperature of ignition, because there is nothing to ignite. As a consequence the injection engine has higher possibilities of efficiency, and those possibilities are attainable, and attained.

Another difference found between the injection and the carburetor engine is that due to the difficulty in making the fuel reach all of the air. With the gas-burning engine and with the carburetor type of volatile-liquid-fuel engine, a mixture is made externally and every part is actively combustible, so that there will be the maximum possible work per cylinder charge for a given compression and shape of combustion line. With the injection engine there are certain real advantages, as pointed out, but if after compression of the air charge the fuel is quickly thrown in, it will be difficult to reach all of the air in an unfavorably shaped combustion chamber from a single point of injection.

There must be a means of spraying the oil into a charge of dense air—air of a density up to 30 or 40 atmospheres—means of arranging to get the injected fuel in contact with as much of the air as possible, and means of preventing the delivery at any point of any considerable amount of fuel that cannot reach air, because in that case carbon will be formed and smoke produced which will choke up the engine in time.

THE TWO CLASSES OF INJECTION ENGINES.

The means of carrying out the operations that are peculiar to the injection engine are divisible functionally into two classes. The air must always be compressed. It may be compressed to ignition temperature and higher, so that the fuel as injected into its ignites immediately and burns as fast as it gets in, the rate of combustion being the rate of injection and controlled by mechanical means. On the other hand, it may be compressed not to ignition but to something less than ignition temperature, and then the fuel injected suddenly to form an explosive mixture which will burn as nearly instantaneously as may be. This gives us two classes of injection engines. The first, the correct operation of which is shown in the indicator card of Fig. 2 in full lines, is explosive in type, and if the combustion is imperfectly carried out there may result explosive shocks as shown at A, or slow burning as at B in dotted lines; the other,

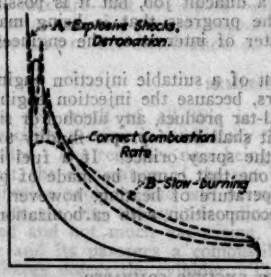


FIG. 2 INJECTION OIL ENGINE—EXPLOSIVE COMBUSTION.

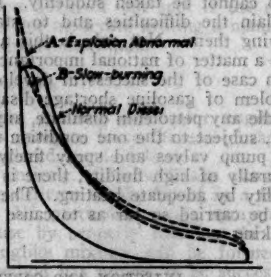


FIG. 3 INJECTION OIL ENGINE—NON-EXPLOSIVE COMBUSTION.

shown in Fig. 3, carries the air to a higher compression pressure, so as to have it not only as hot as the ignition temperature but somewhat hotter. This will give a non-explosive combustion at substantially constant pressure according to the full line, but if combustion be carried out imperfectly, or deranged as to timing of injection or combustion, it may produce explosive shocks as at A or slow burning as at B according to the dotted lines.

Each of these two classes is subject to certain derangements or diseases peculiar to a given mechanism, but comparing the two properly adjusted, how do they stand with reference to each other? Is there any great choice, any reason why the advocate of one should call the advocate of the other wrong? Not at all. Each is justifiable on the grounds of efficiency, power and practicability, so that the real problem boils down to one of mechanical questions of relative cost, reliability, foolproofness and adaptability to service conditions. The two are related in this simple manner as to efficiency. If the compression of the first with explosive-type combustion is about half the compression of the other burning the fuel at substantially constant pressure, their efficiencies are substantially the same. In the former case, if an explosive mixture is to be made, the compression must be kept X degrees below the ignition value in order to keep it under control, and in the other case Y degrees above ignition temperature to insure prompt ignition after injection.

Assuming the exponent n in the equation $PV^n = C$ to have a value of 1.4, it appears that 150 pounds compression will produce the ignition temperature of kerosene (998 degrees F.) if the initial temperature is a little less than 250 degrees F., and if an explosive mixture is to be formed and not preignited, a margin of 100 degrees below ignition will be attained with an initial temperature of something under 200 degrees and a margin of 200 degrees with about 175 degrees F. initial. For fuel oil having an ignition temperature of 1,070 degrees F. the same conditions will be brought about by the same compression when the initial temperatures are 300 degrees F., 250 degrees and 200 degrees respectively.

On the other hand, if the air is to have a safe margin of temperature over the ignition value before compression, higher compression or higher initial temperatures are necessary. For solid-injection sprays it is generally assumed that 200 degrees margin is safe and for air spraying 400 degrees margin. For the latter case a compression of 450 pounds is pretty generally adopted, and this will be secured with a little over 250 degrees initial with kerosene and a little less than 250 degrees with fuel oil. Solid-injection ignition may be produced with equal reliability with less compression, or with lower initial temperature, or both.

METHODS OF SPRAYING FUEL EMPLOYED IN INJECTION ENGINES.

The first mechanical problem in connection with this injection engine is that of making the spray, and one might say, in a way, that the building of the engine begins with the forming of a chamber around a spray. The simplest way of making a spray nozzle, introduced by the first successful commercial engine which was brought here from England—the Hornsby—consists in drilling a hole in a plate. Through this a jet is projected which strikes the wall, a spray being formed by splashing (Fig. 4). If the hole is reduced in size, or if supplied with oil under much higher pressure, the oil will move proportionately more slowly on the sides as compared with the center, and finally the entire jet will expand into a fine mist (Fig. 5).

The essential characteristic of such a spray is strong penetration power. What is necessary, however, is some means of spreading. This can be secured by multiple holes (Fig. 6). It can also be secured by using slots which are normally closed but which are opened by the oil pressure deflecting the metal. Spread can be secured also from a single spray orifice by giving the oil back of the orifice a rotary motion just as in the mechanical atomizer oil burner as developed for the Navy (Fig. 7). Here the oil grooves are arranged to deliver tangentially into a small whirl chamber A, so that the oil in the chamber will have a rotary motion, as also will the

oil issuing from the end of the orifice B at the outlet from the chamber. If the orifice B has an area many times greater than the area through these grooves, there will be no residual pressure in the chamber A, and in this case with sharp edges oil spray will issue in a hollow cone form, due to a pure centrifugal whirl. The spray will have a good uniform

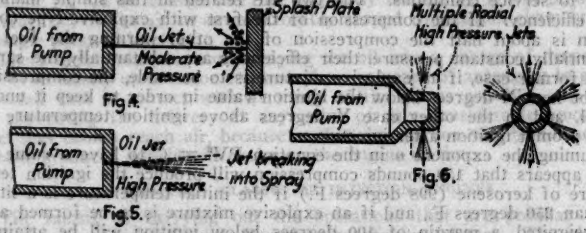


FIG. 4 HORNSBY JET-SPLASH SPRAYING DEVICE.
FIG. 5 HIGH-PRESSURE JET BREAKING INTO SPRAY.
FIG. 6 MULTIPLE RADIAL HIGH-PRESSURE JETS BREAKING INTO SPRAY.

spread with little or no penetration. If, however, orifice B be narrowed down so that it is smaller than the grooves in area, there will be residual pressure in the whirl chamber A, producing axial velocity also, and the cone spray becomes narrower and solid. The spray will then have less spread and more penetration.

In addition to this method of solid-injection spraying there is the air-spray system. This in its simplest form (Fig. 8) consists of a cup with liquid fuel in the bottom, compressed air above the fuel, and a small hole in the side at or below the oil level. As the air escapes through the hole there is first a depression of the liquid right at the point and the liquid is carried to the orifice by the air flow across its surface and blown out, being sprayed by the higher velocity of the air. Such a spray is fine and has good penetration, but not much spread.

A modification giving somewhat better control is shown in Fig. 9, which has a depression in the passageway at A, in which the fuel is deposited as a pool. A deflector over the oil directs the air down and across its surface. The air in motion will tear the liquid off from the surface and this may be delivered to the cylinder in a narrow cone spray through a contracted orifice A, or through multiple orifices as at B, to get an adequate spread. This form is the so-called "open air spray" of the Diesel engine.

Next there is the so-called "closed air-spray valve," in which a valve seats on the outlet of a passage as in Fig. 10, and which when lifted allows the air to flow. The oil is delivered by the pump into the passage and spreads out on plates usually provided with holes, grooves, or slots so as to offer a large amount of surface to be wetted by the oil, which can be blown off gradually and delivered as a spray through a single or multiple orifice A. Such a spray has a narrow angle, strong penetration, and little spread except as may result from impact and rebounding. This is the most common air spray of the Diesel engine. To secure more spread directly, the valve can be reversed in seating as in B.

For any given form of spray, air or solid, there must be a suitable combustion-chamber form, or for any given combustion-chamber form there must be selected a spray of suitable shape or energy to best reach distant air, with always the possibility of setting up turbulence or internal air currents as a corrective means.

DIESEL ENGINES AND THEIR LIMITATIONS.

The first class of engines to be noted under the injection type is the one that is most successfully used commercially—the air-injection Diesel

FIG. 7 SINGLE-ORIFICE WHIRL-CHAMBER SPRAY NOZZLE

FIG. 9 AIR-SPRAY VALVE, OPEN TYPE

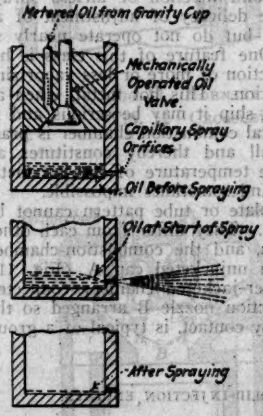


FIG. 8 HVID AIR-SPRAYING CUP.

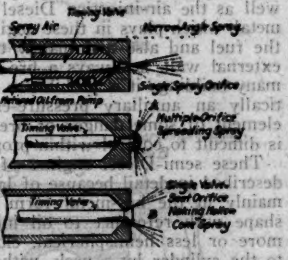


FIG. 10 AIR-SPRAY VALVE, CLOSED TYPE

engine—and which normally has about 450 pounds compression—more than is sufficient to ignite an ordinary oil, enough more to be safely above ignition temperature all the time. Such engines are normally rated at 70 pounds brake mean effective pressure, but are capable of producing over 100 pounds if the metal can stand the intense heating. A fuel consumption of from 0.4 to 0.45 pound per B.H.P. per hour is standard. They are built in all except very small sizes with cylinders up to about 36 inches diameter, depending on speed and mean pressure. The maximum size is limited by the same internal heating conditions with tendency to

crack the metal as obtain for large gas engines. The air-compression Diesel engine is in successful use, as is well known, for both stationary and marine purposes, and is the standard oil engine in use for large ships.

Certain limitations of the air-injection Diesel engine make it unsuitable as a substitute for the gasoline engine. It cannot be made to work with cylinders of too small a size without abnormally high compression, because of the cooling conditions that exist during compression, and in the smallest practical size it is too expensive and too complicated. The control of the air spray is peculiarly delicate. The air for it must be provided by an attached air compressor, and it requires a pressure of never less than 600 and often 1,300 to 1,500 pounds per square inch. Such a compressor small enough for the purposes of an automobile is a mechanical absurdity. The particular field to which the air-injection Diesel is not at all adapted, therefore, is that of the small-cylinder high-speed engine, and that, in the internal-combustion market, is the biggest field of all.

DESIRABLE FEATURES OF SEMI-DIESEL ENGINES.

To approach the problem of the small injection engine, what is available as a starting point? The nearest thing is a type of engine that has been on the market for some time and which is commonly known as the semi-Diesel. These have a feature particularly attractive in the small-engine field—that of operating with so-called "solid-injection or airless spray" that eliminates the air compressor and the delicacy of adjustment of an air-spray valve system. They are simple, but do not operate nearly so well as the air-injection Diesel engine. One feature of this class is hot metal, which plays in these engines the function of more or less vaporizing the fuel and also and mainly that of ignition. This hot metal, when an external wall, is always a fire risk. On a ship it may be serious and in many buildings it is prohibitive. A hot-metal combustion chamber is practically an auxiliary pressure-enclosing wall and thereby constitutes an element of some danger of breakage. The temperature of the hot metal is difficult to control within proper limits, and sometimes impossible.

These semi-Diesel engines of hot-bulb, plate or tube pattern cannot be described in detail because of lack of space. They differ from each other mainly in the hot-metal form, or location, and the combustion-chamber shape with reference to oil injection. An unjacketed cap A (Fig. 11) more or less hemispherical, closing a water-jacketed chamber connected to the cylinder by a neck, with an oil-injection nozzle B arranged so the jet strikes the hot cap to produce ignition by contact, is typical of a group of engines that started with the Hornsby.

COLD-WALL EXPLOSIVE-COMBUSTION SOLID-INJECTION ENGINES.

As a result of a fairly general knowledge of the conditions surrounding these so-called solid-injection semi-Diesel engines, attention has been directed toward substitutes that would have some of the good qualities they had—simplicity, cheapness, foolproofness—as well as the good properties of the Diesel—cold walls, ignition by compression, and cleaner combustion with greater independence of fuel quality. Efforts to produce a cold-wall engine are directed along both lines. One is the explosive-combustion engine; the other, the non-explosive-combustion engine. Cold-wall explosive-combustion solid-injection engines are comparatively new. The British Crossley (Fig. 12) has a piston with a conical end and a cylindrical projection A. The cylinder head is completely water-jacketed. As the piston approaches the head, the projection A will pass the corner

of the cylinder head, at which time the air in the annular space B is trapped. This projection is a loose fit, but not too loose, so that during the time it is passing into the head bore there is a violent annular stream of hot compressed air cylindrically distributed down the sides of the combustion chamber and back along the center. Into that stream of air is injected a fine spray of oil that is instantaneously ignited, burning as fast as oil and air come together. The combustion is explosive in type but not in fact.

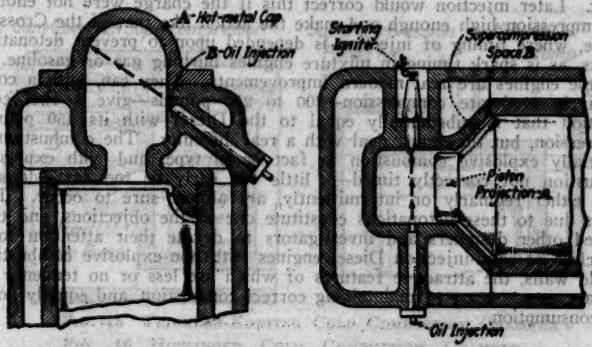


FIG. 11 HORNSBY TYPE OF COMBUSTION CHAMBER.

FIG. 12 CROSSLEY COLD COMBUSTION CHAMBER.

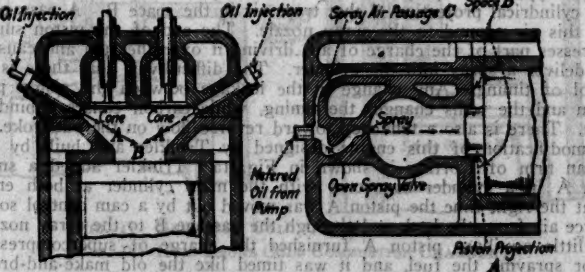


FIG. 13 PRICE-INGERSOLL-RAND-DE LA VERGNE COLD COMBUSTION CHAMBER

FIG. 14 HASELWANDER COLD COMBUSTION CHAMBER.

An American representative of the same class of engine is found in the Price construction (Fig. 13), which works differently and in which the combustion is normally explosive in fact. Here a fine spray is injected into a conical combustion chamber A on each side of a central cylindrical chamber B, up which a gentle air current rises during compression. This

serves to help mix the fine spray with the air during the last of the compression stroke. Compression is adjusted so that the ignition temperature is reached just before the end, and as the air charge before compression is cooler at no load than at full load, the misfires that would happen, due to failure to reach ignition temperature, are prevented by an air throttle having the effect of retaining enough hot burnt products to avoid misfires. Should the air charge get too hot from any cause, the whole charge might be ignited everywhere at the same time before compression was complete, producing a detonating combustion with explosive shock. Later injection would correct this if the charge were hot enough or compression high enough and make the action like that of the Crossley engine, where timing of injection is depended upon to prevent detonating shocks, as is spark timing in mixture engines burning gas or gasoline.

These engines are real modern improvements. They can, with a comparatively moderate compression—200 to 250 pounds—give a fuel consumption that is substantially equal to the Diesel with its 450 pounds compression, but they must deal with a real difficulty. The combustion is essentially explosive combustion in fact or in type, and with explosive combustion not correctly timed—a little too early or too fast—detonations, either regularly or intermittently, are almost sure to occur. The shocks due to these detonations constitute one of the objections, and this has led other designers and investigators to devote their attention to a new class of solid-injection Diesel engines with non-explosive combustion in cold walls, the attractive features of which are less or no tendency to detonate, greater ease of maintaining correct combustion, and equally good fuel consumption.

SOLID-INJECTION DIESEL ENGINES WITH NON-EXPLOSIVE COMBUSTION IN COLD WALLS.

An early attempt to eliminate the air compressor from Diesel engines was made by the German Haselwander, as shown in Fig. 14, which is practically a Diesel engine with air injection, but without a compressor. An open type of air spray is combined with a piston construction embodying a cylindrical projection A that traps air in the space B. A passage C leads this air around to the spray nozzle. The end of the piston supercompresses part of the charge of air, driving it over the fuel and causing it to deliver a spray into the cylinder. The difficulty is that there is no control of timing. Any change in the leakage between the piston projection and the walls changes the timing, while carbon causes a binding action. There is also a tendency toward reverse flow on the outstroke.

A modification of this engine designed by Trinkler and built by the German firm of Körting is shown in Fig. 15. Trinkler added a small piston A in a cylinder connected with the main cylinder at both ends. Just at the right time the piston A was moved out by a cam control so as to force air from the back end through the passage B to the spray nozzle. This little auxiliary piston A furnished the charge of supercompressed air for spraying the fuel, and it was timed like the old make-and-break ignition of the gas engine. The objection to it was that it tended to stick and stop the engine.

In Höflinger's proposed engine, Fig. 16, a small cuplike cylinder with a piston projects into the working cylinder. A passage for oil is connected below the piston and before the fuel is wanted it is deposited by the pump in that cup or bottom end of the small cylinder. Just when it is wanted a timing cam drives the piston down, compressing air on top of the fuel charge and expelling both through holes A in the side as an air spray. So far as is known, this engine has not been built, but it is

very suggestive, especially when considered along with the very modern one reported from Detroit by Gernandt, Fig. 17, with special reference to use for automobiles. This has the Höflinger cup and fuel feed, but with a connection to a timing valve and to an auxiliary cylinder with a piston. The piston of Höflinger's cup is replaced by an independent piston A and separate timing valve B. Then at the right time the piston descends, the valve opens by a cam movement and the charge of air is supplied to the cup, spraying the oil into the cylinder.

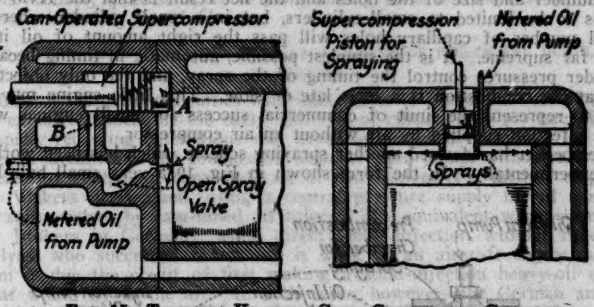


FIG. 15 TRINKLER-KORTING COLD COMBUSTION CHAMBER.

FIG. 16 HOFLINGER COLD COMBUSTION CHAMBER.

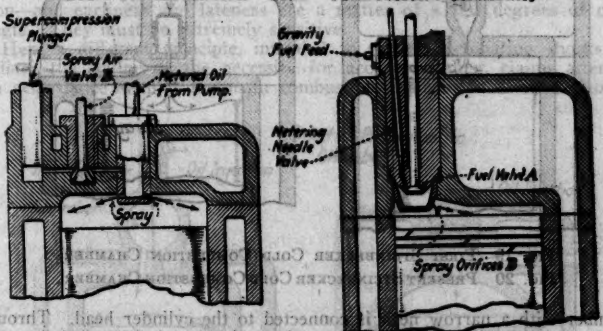


FIG. 17 GERNANDT COLD COMBUSTION CHAMBER.

FIG. 18 HVID-BRONS COLD COMBUSTION CHAMBER.

Finally, the simplest of all in this class and the one form that has come into almost universal use for small stationary engines, and has practically no competition in its own field—farm units—is the Hvid, shown in Fig. 18. The Hvid retains the same fuel cup as the last two engines, but with entirely different connections for fuel and provisions for supplying and timing the spraying air. The fuel is delivered into the cup by gravity through a mechanical valve A, so that after it is in the cup the latter is a closed chamber. The fuel is delivered into the cup long before injection, in fact, before compression begins. During compression air flows

into the cup through three holes, B, which prevent the outflow of oil prematurely, provided the holes are small enough so as to be capillary. After compression is complete, the air in the cup has a lower pressure than the air in the cylinder, which differential is quickly equalized on the expansion stroke. As soon as the cup pressure exceeds the cylinder pressure, the cup air will spray the oil into the cylinder. If the holes are too large, fuel will leak into the cylinder while the cup is being charged. If they are too many, the air escaping into the cylinder will come out of some without spraying oil out of the others. There is a limit to both the number and size of the holes and the net result is that the Hvid cup seems to be limited to small cylinders, but in a small cylinder where a small number of capillary holes will pass the right amount of oil it is thus far supreme. It is the simplest possible, automatic in timing because cylinder pressures control the timing of the spray, and the only objection is that the combustion is a little late or slow. This little engine may be said to represent the limit of commercial success for small engines with solid injection supplied by air without an air compressor.

Steinbecker has devised another spraying scheme which has been worked out experimentally. In the form shown in Fig. 19 a very small bulb-like

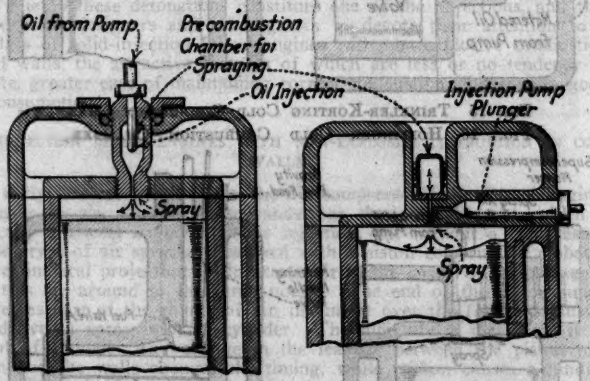


FIG. 19 FIRST STEINBECKER COLD COMBUSTION CHAMBER.
FIG. 20 PRESENT STEINBECKER COLD COMBUSTION CHAMBER.

chamber with a narrow neck is connected to the cylinder head. Through the top of this chamber a spray nozzle projects and delivers a jet or a coarse spray into the neck. At this time the air has already been compressed to and above the ignition temperature, so that as the oil escapes it immediately ignites, but not much can burn because not much of it can reach air in the narrow passage. What burns on the back face of the spray, according to Steinbecker, will raise the pressure in the bulb so as to produce an outflow of gases to respray finely what oil has been delivered to the neck and deposited on its walls. By the rise of pressure in the bulb, Steinbecker expected to really spray the main charge of oil.

Steinbecker has a recent engine, Fig. 20, in which the bulb chamber is retained, the neck passage is made extremely narrow so as to get a very high velocity through it, and injection pump delivery is led into the middle of the neck. It is timed so that oil enters near the end of the

compression stroke while there is still some upward flow of air from the cylinder to carry it into the bulb chamber, which is unjacketed and hot, so as to produce ignition and combustion by explosion. The hot gases produced cause a reverse flow to the cylinder and spray the oil delivered later directly into the cylinder. This is Steinbecker's engine as it now stands, and as operated experimentally even in automobiles. The timing is directly by the pump, and should that pump force the oil in too soon, there would be an explosive shock. If the timing were wrong, the correct amount of oil would not enter the bulb and the spraying would fail also. It is probably quite sensitive to pump timing, as is not the case with some others, including the Hvid.

THE DIVIDED COMBUSTION CHAMBER AS A MEANS OF PREVENTING DETONATING SHOCKS.

One of the great difficulties with injection is to prevent the development of explosive shocks by too early, and loss of efficiency by too late, timing. Rapidly recurring detonations will wreck any machine in time. To direct the oil stream into the combustion chamber by a pump, without any other means of control of time and rate of combustion, is a method employed by Vickers in England, using a central pressure supply of oil admitted to spray valves by cam-timed oil valves. The equivalent was worked out by Junkers in Germany, using direct pump injection without timed oil valves, who succeeded in making it work in an aircraft engine, and to him is due the credit of first making a solid-injection heavy-oil engine that would fly in the air. In both cases, however—the German and the British—the fuel went directly into the cylinder and the production of detonating shocks was entirely a question of avoiding too early an injection—and earliness and lateness are a matter of a few degrees of crank angle. They must be extremely sensitive.

Here a different principle, intended to prevent detonating shocks and relieve the engine of the necessity for accurate timing, claims attention. In accordance with it the main combustion chamber is divided into two

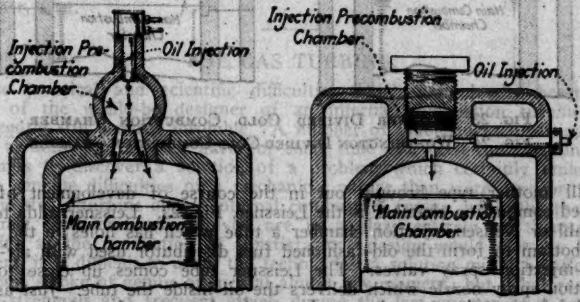


FIG. 21 NYDAHL DIVIDED COLD COMBUSTION CHAMBER.
FIG. 22 NIELSEN DIVIDED COLD COMBUSTION CHAMBER.

parts, one in the cylinder and the other removed—a divided combustion chamber. The air is compressed partly in the cylinder and partly in a connecting pocket, the major part in the former.

The first construction embodying this principle to be noted here is a Scandinavian one by Nydahl, shown in Fig. 21. He provided a spherical

chamber connected to a cylinder by a series of holes and with an oil spray at the top. The piston stops at a point that leaves considerable air in the cylinder. There is a body of air in the cylinder and another body in the injection chamber at dead center. If the quantity of air in the injection chamber is small enough in comparison with that in the cylinder, then when the fuel is injected—the whole charge of fuel into the small amount of air—it cannot produce any explosive shock because so small an amount can burn. What, then, will happen to the rest of the fuel? It will change as it would in a gas producer where fuel reacts with less air than is required for combustion. Some of it will burn; the rest will gasify; possibly some will merely vaporize. In this divided-combustion-chamber construction the piston becomes the principal element of combustion timing, somewhat as in the Hvid engine because the main combustion is produced by the flow from injection chamber to cylinder. The divided combustion chamber is the principal element in preventing the explosive shocks, the piston movement controls the completion of combustion, and if the compression is high enough the whole structure can be jacketed.

Such a water-jacketed injection chamber, but of more or less cylindrical form, and embodying the divided combustion chamber and side injection of fuel, is shown in Fig. 22. This is a Danish form, by Nielsen, who also provided for adjustment of injection-chamber volume by a special cover or cap.

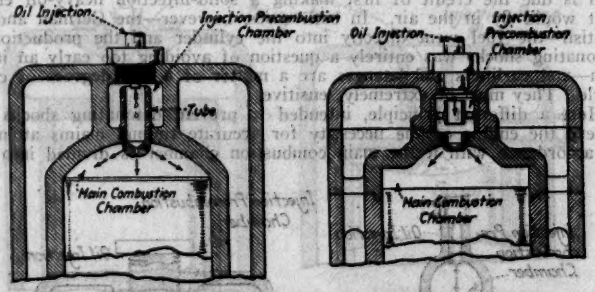


FIG. 23 LEISSNER DIVIDED COLD COMBUSTION CHAMBER.
FIG. 24 WORTHINGTON DIVIDED COLD COMBUSTION CHAMBER.

Still another type brought out in the course of development of the divided combustion chamber is the Leissner, Fig. 23. Leissner adds to the Nydahl or Neilsen injection chamber a tube which has holes in the sides and bottom to form the old-fashioned fuel distributor used with air- and solid-injection spray valves. The Leissner tube comes up close to the injection spray nozzle which delivers the oil inside the tube. Just as the small injection chamber alone prevents development of explosive shocks by limited contact of air and fuel, so does insertion of this tube add something to further limit the contact control. Leissner specifies that the holes through the bottom and sides of the tube, and the space above it, shall be so related to each other in area as to produce the following series of actions: Compression carried first to ignition temperature; injection inside the tube in the injection chamber and partial combustion in the tube, producing a rise of pressure in the tube, which in turn projects jets

of still unburned oil sidewise into the air around the tube. Combustion of these jets raises the pressure outside of the tube and causes reversal of flow back into the tube and down through it to the cylinder, helped by the movement of the piston; the air left around the tube finally passing through the tube and expelling the fuel charge in front of it into the cylinder, the space around the tube and in the tube being in series. This is a new engine brought out in Sweden, and it is now being introduced into this country.

Finally, in this class of divided-combustion-chamber injection engines there is a new one developed by Worthington—the class in which the injection or precombustion chamber is used to limit the development of explosive shocks and connected to the cylinder by an ejection orifice through which gasified fuel is expelled to the cylinder by a pressure differential due partly to the movement of the piston and partly to the precombustion. This is shown in Fig. 24. It has a tube open at both ends and supported from the side walls with wide spaces at each end. The bottom of the injection chamber has a passage to the cylinder of the usual fuel-distributing sort, forming a fuel-ejection orifice. There are large holes through the tube-supporting web, so that the injection chamber acts as a single chamber, with the tube acting as a sort of fuel baffle or guard. The charge being injected into the tube is limited by the tube as to air contact. The tube constitutes a fuel guard, by means of which the amount of precombustion can be controlled and at the same time the fuel is gasified in the hot limited amount of air. A rise of pressure in the injection chamber, due to partial combustion, is supplemented by depression of the pressure on the cylinder side due to the piston movement, which starts the flow of parallel streams of air and fuel toward the cylinder. This action creates in effect a bunsen burner in the top of the cylinder. This arrangement gives a construction practically as simple and foolproof as the little Hvid, except for the injection pump, but one that seems to be adapted to a wider range of cylinder sizes and speeds and less limited by fuel quality.

Much experimental research has already been completed and more is planned, to be presented in a later paper.—"Mechanical Engineering," October, 1921.

THE GAS TURBINE.

The practical and scientific difficulties which undoubtedly beset the path of the would-be designer of an internal combustion turbine are apparently still proving insoluble. A number of independent investigators, both at home and abroad, have for some time past been concentrating on attempts to discover a solution of a problem which certainly embraces more genuine difficulties of a mechanical nature than were ever provided by the steam turbine or the internal combustion engine, while the thermodynamic considerations involved are in a class entirely by themselves. Baffling as the production of a satisfactory design would seem to be at present, it is difficult to avoid the feeling that a solution might be afforded by some stroke of genius or luck at almost any moment; but it has resisted such strenuous and frequent attack during the last quarter of a century with such success that the probability of an early realization of an efficient internal combustion rotary engine seems to be rather on the wane than otherwise, and some at least of those who have gone very deeply into the matter during the last two or three years are reluctantly abandoning their efforts. Experience of the steam turbine has taught a great deal that is of value in approaching the internal combustion turbine, particularly in the direction of its limitations. Similarly, experience with petrol or large Diesel engines has indicated where difficulties are apt to arise in

using other forms of energy. But in attacking a problem such as that of a gas combustion turbine, it is absolutely impracticable to be bound down by any conventional ideas in regard to design beyond the fundamental thermal consideration of heat received minus that rejected, in its ratio to that received, which applies alike to all heat engines.

To obtain combustion energy in a utilizable form, whether for expanding through a nozzle and yielding up kinetic energy in a jet, or whether the static expansion force behind a piston is to be used as in a Diesel engine, compression is required, and here the mechanical difficulties at once begin. Good results can be obtained with rotary air compressors only up to comparatively low pressures; 100 pounds per square inch is about the limit yet obtained, and it involves the use of a large number of fans in series. We know that if it were possible to solve the problems at the high-pressure end of the gas turbine we could utilize a considerably lower exhaust temperature than is possible in a Diesel engine, just as one can efficiently use lower vacua in steam turbines than in low-pressure cylinders; and consequently, to attain the same thermal efficiency the initial pressure and temperature need not be so high, but the compression will require to be of an order quite unattainable with rotary compressors. Hitherto, in all the gas turbines tried, this negative work of compression has practically nullified the positive work of the turbine itself, and in cases in which the compression has been provided externally the power required has usually even exceeded the turbine output. Another objection to direct-driven rotary compressors lies in the necessity of meeting the conditions of starting up or of altering the revolutions of the main turbine. At the cost of adding to the negative work in friction a reciprocating compressor will be essential to overcome this difficulty. The turbine will require to be operated on a continuous combustion system, as no fluctuation of rotary speed even as small as that found in the Diesel engine will be satisfactory. By splitting the power up into eight cylinders the volume of oil introduced into each on every alternate stroke is reduced to a very small quantity, and though at 120 revolutions the working stroke occupies but one-quarter of a second, the problem of dissipating the heat absorbed by the metal of the cylinder and particularly by the cylinder cover is a formidable one, and quite as much of thermal energy in the fuel as is converted into work goes overboard in the cooling water. This cooling is continuous, and yet the explosion stroke only occupies one-quarter of the time. How is a combustion chamber that is to be subjected to continuous heat likely to fare? Here we come across the germs of a wide series of problems relating to metals under the influence of heat. A boiler furnace stands up to 2,500 degrees F. for years on end without difficulty, but its function is to transmit the heat as fast as it can; but the function of the turbine material is to withstand pressure and not to pass more than is absolutely necessary to avoid rupture due to expansion strains. It is possible to keep this combustion small by subdividing the amount of fuel supplied over a number of separate chambers, and further over a number of turbine wheels in parallel, and by arranging the firing so that a time element is introduced to assist in the heat transmission from each chamber. No scheme that offers itself seems able to meet the double requirement of adequately cooling the combustion chamber and of reducing the existing loss of heat carried away in the process. The next difficulty which requires to be circumvented—space forbids the mention of more than a few—lies in the fact that the efflux velocity of high-pressure and high-temperature gas expanding to a lower pressure is very considerable. There has been a general tendency in steam work to reduce velocity through nozzles to a point, where possible, below the critical value so that the use of divergent nozzles is obviated. This is

practically impossible in internal combustion work, and the uncertainty of the proper proportions to provide for the nozzles is a further source of inefficiency, though the real objection lies in the difficulty of suitably harmonizing wheel speed and jet speed when the latter is high. Actual high speed of revolution is immaterial; gearing corrects that. But it is quite impossible to exceed certain peripheral speeds depending on the form of wheel and blade attachment, and *with stiff shafts* a limit in the neighborhood of 500 feet per second is involved. For maximum efficiency and a single row of blades this limits the gas speed to far below its actual value; velocity compounding must therefore be resorted to, and even then a sacrifice will be made. Thus we find that the fundamental condition on which efficiency depends—that is, on the transference of the kinetic energy to the turbine wheel—is, step by step, being reduced to a position in which the quantity of energy transferred minus the negative work expended in compression will not reach the standard of thermal efficiency obtained in the reciprocating unit. The largest gas turbine unit so far constructed has, we believe, not attained more than about a quarter of its designed 1,000-kilowatt output. The early efforts of Armengaud and Lemale in France and of Holzwarth in Germany did little else than indicate the wide extent of the difficulties involved in dealing with a fluid other than steam or water in rotary engines. Quite a number of experiments have been conducted in this country and in Germany, but in all cases the negative work of compression has proved to be the stumbling block. This fact in itself indicates the attainment of a very low turbine efficiency, because the work of driving the air compressor for a big Diesel engine for charging purposes is only of the order of about 5 per cent of the full power, and, from a consideration of the other losses, that absorbed on the compression stroke cannot be very great. We have not in the foregoing sketch of the pure gas turbine referred to mixed combustion turbines. The possibilities of water injection into the combustion chamber, with the double object of reducing its temperature, and in doing so utilizing the potential energy thus transferred in the fluid impinging on the wheel instead of letting it flow uselessly away in the cooling water, have not been overlooked, nor has the use of the exhaust gases, somewhat on the lines of the Still engine, been ignored. In lieu of using them to raise steam, it has been proposed to pass them through a turbine, but the fallacy of the idea soon becomes apparent on investigation. The injection method, however, offers several possibilities that the pure combustion turbine does not, and it is along these lines that some development may be eventually expected.

The complete solution of the problem would be a step of the most profound importance in engineering. The fact that inside one large cylindrical casing it would be possible to transmute the thermal value of the liquid fuel in the bunkers into brake horsepower on the shaft without the mass of separate intermediate parts which constitute the existing engine would have an extraordinarily far-reaching effect on every branch of manufacture and power supply. Ideal as the suggestion may be, or nebulous as the prospects of its realization seem to be at present, the introduction of a motor which dispensed with steam generating and condensing plants and was applicable in all sizes from motor cars to mammoth liners would create a veritable upheaval in all sections of industry. We really wonder how many firms which would be very intimately affected by such an occurrence have carefully considered what their position would be in the event of such a motor being forthcoming without much warning. That it would entirely supersede the use of steam machinery is most unlikely, but for the larger and more important power plants and steamships no motor could offer a more ideal form

of prime mover. Pleasant as it may be to speculate on the presence of such perfection, the difficulties in the way of its attainment have hitherto proved too formidable. They are not, however, provably impossible in theory, and consequently it is within the bounds of possibility that such a motor may be forthcoming, and, if so, it is not unlikely to take a number of people by surprise.—"The Engineer," September 23, 1921.

THE CALCULATION OF CRITICAL SPEEDS.

In a notable paper, which constituted, in fact, a veritable mine of valuable information, contributed to the Proceedings of the Institution of Electrical Engineers in 1912, Mr. K. Baumann gave a rule for the critical speed of turbine rotors, which, from the context, appears to have been arrived at empirically. He said that if the maximum deflection δ of the rotor under gravity were calculated, then it had been found by actual trial for a large variety of rotors that the critical speed was given by the relation

$$\delta\,\omega^2 = (1.07 \text{ to } 1.08)\,g.$$

In this expression ω denotes the angular velocity in radians and g the acceleration due to gravity. Naturally an empirical rule of this kind cannot be expected to apply to all types of rotor, but it is possible to obtain nearly as simple an expression applicable in all cases, by making use of the late Lord Rayleigh's method of estimating the natural periodicity of vibration of a loaded shaft; which is, as is well known, identical with the critical speed.

The formula is given in most textbooks on the steam turbine and may be put in the form

$$\omega^2 = \frac{386.4 \int_0^l p\,\delta\,dx}{\int_0^l p\,\delta^2\,dx}$$

In this expression p denotes the load per inch run at any point of the rotor, and δ the deflection (under gravity) of the rotor at the same point; whilst l is the total length of the rotor between centers of bearings and 386.4 is the value of gravity in inches per sec.^{2} In practice it is a very tedious operation to evaluate these integrals if treated in the usual way, but a great simplification can be effected by making use of the relation

$$p = \frac{dS}{dx}$$

where S denotes the shearing force at any point of the rotor, which for this purpose, may be most conveniently dealt with as if supported at one end only, cantilever fashion.

If, in the integrals, we now substitute dS for $p\,dx$ we get

$$\omega^2 = \frac{386.4 \int_0^{S_1} \delta\,dS}{\int_0^{S_1} \delta^2 dS} = \frac{386.4 \;.\; S_1 \text{ (mean value of } \delta)}{S_1 \text{(mean value of } \delta^2)}$$

Cancelling out S_1 from top and bottom, we get simply

$$\omega^2 = 386.4 \frac{\text{(mean value of } \delta)}{\text{(mean value of } \delta^2)}$$

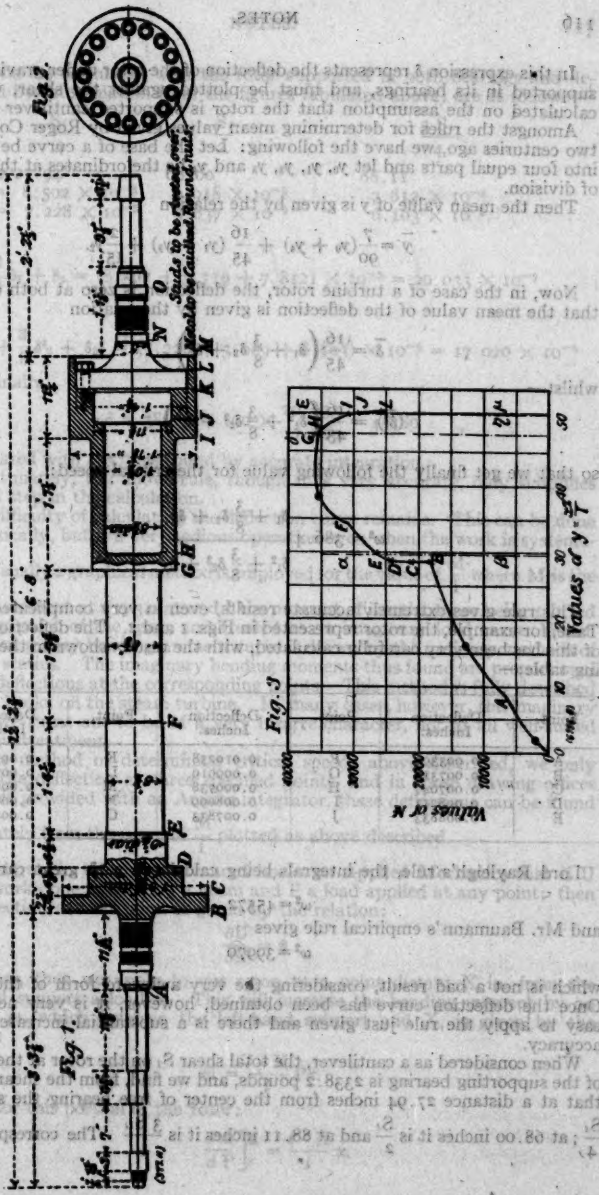


Fig. 2.
Studs to be riveted over
Studs to be Cadmium Round nuts
N O
I J K L M
G H
F
E
D
C B
A
Fig. 1
Fig. 3
Values of N
Values of y = 1/20 y

In this expression δ represents the deflection of the rotor under gravity when supported in its bearings, and must be plotted against the shear, which is calculated on the assumption that the rotor is supported cantilever fashion.

Amongst the rules for determining mean values given by Roger Cotes over two centuries ago, we have the following: Let the base of a curve be divided into four equal parts and let y_0, y_1, y_2, y_3 and y_4 be the ordinates at the points of division.

Then the mean value of y is given by the relation

$$\bar{y} = \frac{7}{90}(y_0 + y_4) + \frac{16}{45}(y_1 + y_3) + \frac{2}{15}y_2.$$

Now, in the case of a turbine rotor, the deflection is zero at both ends, so that the mean value of the deflection is given by the relation

$$\bar{\delta} = \frac{16}{45}\left(\delta_1 + \frac{3}{8}\delta_2 + \delta_3\right)$$

whilst—

$$(\overline{\delta^2}) = \frac{16}{45}\left(\delta_1^2 + \frac{3}{8}\delta_2^2 + \delta_3^2\right)$$

so that we get finally the following value for the critical speed:

$$\omega^2 = 386.4\,\frac{\delta_1 + \frac{3}{8}\delta_2 + \delta_3}{\delta_1^2 + \frac{3}{8}\delta_2^2 + \delta_3^2}$$

This rule gives extremely accurate results, even in very complicated cases. Take, for example, the rotor represented in Figs. 1 and 2. The deflection curve of this has been very carefully calculated, with the results shown in the following table:—

Point.	Deflection Inches.	Point.	Deflection Inches.	Point.	Deflection Inches.
A	0.003399	F	0.010238	K	0.007294
B	0.007313	G	0.009919	L	0.007023
C	0.007647	H	0.009738	M	0.006832
D	0.008211	I	0.008099	N	0.006459
E	0.008833	J	0.007833	O	0.002692

Lord Rayleigh's rule, the integrals being calculated with great care, gave

$$\omega^2 = 45572$$

and Mr. Baumann's empirical rule gives

$$\omega^2 = 39970$$

which is not a bad result, considering the very awkward form of the rotor. Once the deflection curve has been obtained, however, it is very nearly as easy to apply the rule just given and there is a substantial increase in the accuracy.

When considered as a cantilever, the total shear S_1 on the rotor at the center of the supporting bearing is 2338.2 pounds, and we find, from the shear curve, that at a distance 27.94 inches from the center of free bearing the shear is $\frac{S_1}{4}$; at 68.00 inches it is $\frac{S_1}{2}$ and at 88.11 inches it is $\frac{3}{4}\frac{S_1}{}$. The corresponding

deflections (as calculated with the rotor supported in both bearings and determined by interpolation from the figures tabulated above) are as follows:—

Shear.	$\frac{S_1}{4}$	$\frac{S_1}{2}$	$\frac{3\,S_1}{4}$
Distance	27.94″	68.00″	88.11
δ =	8.502×10^{-3}	9.918×10^{-3}	7.812×10^{-3}
δ^2 =	7.228×10^{-5}	9.837×10^{-5}	6.103×10^{-5}

Hence

$$\delta_1 + \frac{3}{8}\delta_2 + \delta_3 = (8.502 + 3.719 + 7.812) \times 10^{-3} = 20.033 \times 10^{-3}$$

and

$$\delta_1^2 + \frac{3}{8}\delta_2^2 + \delta_3^2 = (7.228 + 3.689 + 6.103) \times 10^{-5} = 17.020 \times 10^{-5}$$

Hence finally

$$\omega^2 = 386.4 \times 100 \times \frac{20.033}{17.020} = 45480$$

as compared with 45572 as found by accurate integration.

Unfortunately, the above rule, though it is very accurate, only simplifies the final step in the calculation.

The difficulty of calculating the deflection curve remains. This can be done arithmetically, but is a very tedious operation even when the work is systematized. Usually a graphical method is employed for the value of $\frac{M}{I}$ where M is the bending moment at any point and I the moment of inertia, these being plotted for every point of the span, and treated as an imaginary load curve. From this an imaginary bending moment curve is drawn by the ordinary methods of graphic statics. The imaginary bending moments thus found are proportional to the deflections at the corresponding points. This method is fully described in most books on the steam turbine. In many cases, however, the imaginary load curve turns out to be of a very bizarre character, not at all well-suited to graphic treatment.

In the method of determining critical speeds above described, we only require the deflection at three specified points, and in those drawing offices which are provided with an Amsler integrator, these deflections can be found immediately from the curve of $\frac{M}{I}$ plotted as above described.

It is shown in the more advanced books on Applied Mechanics that if U be the work done in bending a beam and P a load applied at any point; then the deflection at that point is given by the relation:

$$\frac{\partial U}{\partial P} = \delta$$

Suppose, then we apply to our rotor, at a point distant X in. from the left-hand support a load P. This will increase the bending moment by an amount m, which between the left-hand support and X, is given by the relation:

$$m = \frac{l - X}{l} Px.$$

so that for this portion of the rotor:

$$\left[\frac{dm}{dP}\right]_0^X = \frac{l - X}{l} x.$$

Similarly, if distances from the right-hand support are denoted by v, so that the point of application of the load is V in. distant from the right support, we get:

$$\frac{dm}{dP}\Big]_0^V = \frac{l-V}{l}v. = \frac{X}{l}.v.$$

Now if N denote the bending moment at any point of a beam, the work done in bending it is:

$$U = \int_0^l \frac{N^2}{2EI}dx = \int_0^X \frac{N^2}{2EI}dx + \int_0^V \frac{N^2}{2EI}dv$$

where, of course, we have

$$V + X = l = v + x.$$

Now if M be the moment due to the weight of the rotor and m that due to the additional load applied at X, we have:

$$N = M + m \text{ and}$$

$$U = \int_0^X \frac{(M+m)^2}{2EI}dx + \int_0^V \frac{(M+m)^2}{2EI}dv$$

so that

$$\delta_x = \frac{dU}{dP}$$

$$= \int_0^X \frac{M+m}{EI}\cdot\frac{dm}{dP}\cdot dx + \int_0^V \frac{M+m}{EI}\cdot\frac{dm}{dP}\cdot dv$$

$$= \frac{V}{l}\cdot\int_0^X \frac{M+m}{EI}x\,dx + \frac{X}{l}\int_0^V \frac{M+m}{EI}v\,dv$$

Now this expression holds whatever the value of P, and hence it will still hold if P = o, in which case m is also zero.

Hence, finally, we get for the deflection at the point X of the rotor, under its own load, the expression:

$$\delta_x = \frac{V}{l}\cdot\int_0^X \frac{M}{EI}x\,dx + \frac{X}{l}\int_0^V \frac{M}{EI}v\,dv \quad \text{. . . . . (1)}$$

If $\frac{M}{I}$ be plotted for each point of the span we can obtain both of these integrals directly by means of the Amsler integrator. Unfortunately, this is an expensive instrument, and few offices are equipped with it. Where available, however, it provides in conjunction with the formula previously given much the simplest method of determining the critical speed of a complicated rotor.

Theoretically, if instead of plotting $\frac{M}{I}$ against x, we plotted it against x^2, we could determine the value of the above integrals by the ordinary planimeter, but the curve thus obtained is, with such a rotor as is represented in Figs. 1 and 2, of an impracticably awkward character.

We can, however, proceed in another way, and thus obtain the value of the integrals by the use of the planimeter only. In general the value of I changes abruptly, but is constant for any particular section of a rotor. Thus for the rotor illustrated in Fig. 1 we may write:

$$\int_0^x \frac{M\,x}{I}\,dx = \frac{1}{I_A}\int_0^A M\,x\,dx + \frac{1}{I_B}\int_A^B M\,x\,dx + \text{etc.}$$

Let $y = \frac{x^2}{I}$, then $dy = 2\,x\,dx$.

so that

$$\int_0^x \frac{M\,x}{I}\,dx = \tfrac{1}{2}\int_0^A M\,dy + \tfrac{1}{2}\int_A^B M\,dy + \text{, etc.}$$

$$= \tfrac{1}{2}\left[\frac{x_A^2 - x_0^2}{I_A} M_A + \frac{x_B^2 - x_A^2}{I_B} M_A + \text{, etc.}\right]$$

Where M_A denotes the mean value of M between o and A, and M_B the mean value between A and B.

Hence if M be plotted against y, areas of the curve will represent twice the corresponding integral

$$\int_0^x \frac{M\,x}{I}\,dx \text{ ,}$$

We proceed as illustrated in the annexed table, the rotor being that represented in Fig. 1 *ante*.

Section.	x.	$100\,x^2$.	$\Delta\,100\,x^2$.	I.	$\Delta\frac{100\,x^2}{I.} = 100\,\Delta\,y$.	$100\,y = 100\,\Sigma\Delta y$.
O A	5·813	3,379	3,379	3·976	849·8	849·8
A B	16·969	28,798	25,419	12·565	2023·0	2872·8
B C	19·407	37,675	8,877	10,492	0·9	2873·7
C D	23·532	55,366	17,691	17,553	10·1	2883·8
D E	28·282	79,976	24,610	30·582	80·5	2964·3
E F	44·345	196,604	116,628	227·38	514·0	3478·3
F G	65·97	435,204	238,600	188·79	1270·0	4748·3
G H	68·22	465,497	30,293	1106·0	27·3	4775·6
H I	85·22	726,245	260,748	849·8	307·0	5082·6
I S	87·72	769,450	43,235	5812·0	7·51	5090·1
J L	92·47	855,070	85,500	5132·7	16·7	5106·8

In compiling this table the values of x^2 are taken from Barlow's tables. The fourth column is the difference between successive numbers in the third column, and these differences divided by I, give $\Delta\,y$ as tabulated in column 6, and each entry in the seventh column is the sum of the preceding numbers in the sixth column, and gives the value of y against which has to be plotted the

corresponding values of M. The curve thus obtained is shown in Fig. 3. Where the intervals between successive values are large as between O and A; A and B; F and G; intermediate values of y and M are calculated, but this is not necessary elsewhere. In this curve the area up to the vertical line $\alpha\beta$ represents twice the integral

$$\int \frac{M x}{I} . d x$$

between o and $x = 27.94$ in., this being the point at which the shear on the rotor if supported at the right hand end only would be one-quarter of the total shear. Similarly the area of the curve up to the vertical $\gamma\eta$ gives twice the value of the above integral between o and $x = 68$, whilst the area up the third vertical $\epsilon\mu$ gives twice the value of the integral between o and $x = 88.11$ in.

The value of the integral $\int \frac{M v d v}{I}$ is found in a similar way, distances from the right hand support being tabulated instead of the left-hand support. It is not necessary in either case to carry the tabulation beyond some three-fourths of the span.

From these integrals the deflection at the points in question are then obtained by equation (I) *ante*.

Where an Amsler integrater is available it is, however, preferable to use it, as rather less preliminary arithmetic is required.

Naturally in calculating critical speeds, any other of the formulæ for mean values given by Cotes can be used. Thus if we divide the shear into three equal parts instead of four, and calculate the deflections at the two intermediate points of division, we get

$$\omega^2 = 386.4 \frac{\delta_1 + \delta_2}{\delta_1^2 + \delta_2^2}$$

but the possible error made is, of course, smaller the greater the number of deflections used.—"Engineering," July 1, 1921.

ANGULAR VIBRATIONS IN MARINE PROPELLING MACHINERY.

By Richard Gardner.

The existence of torsional vibration of the propeller shafting in ships driven by reciprocating engines is well known; where such vibrations have occurred under conditions of synchronism, what in acoustics is called resonance, the effects have sometimes been disastrous. This vibration effect in relation to geared steam turbine installations has been very little investigated; but it is believed at the present time to be a cause, in many cases, of troubles which have not been satisfactorily explained otherwise.

A mechanical vibrating system consists of flexible parts and inert parts, and during vibration the motion of all the elements is such that some point or node is maintained in equilibrium under the forces about it. (There may be several nodes.) If the flexibility of the system is confined to the torsional flexibility of the shafting between the propeller and the gear wheel, then the latter cannot be a node or equilibrium point, but must, in some degree, be subject to vibration.

The angular vibration of the gear wheel may be so small in amplitude that the only result is a variation of pressure on the teeth, and there is no reason to suppose that this would damage the teeth or even detract from smooth running.

A greater amplitude would cause the teeth to come out of contact, and the condition of the teeth in some cases after the gear has run for a short time gives a strong impression that separation and striking of the teeth has actually occurred.

It is notable that the unsatisfactory features found in experience with double reduction gearing were practically unknown so long as gearing was confined to the single reduction type. In order to show a comparison of the two types in relation to the effect of torsional vibrations, let the condition be written down that separation cannot occur between the gear wheel teeth and the teeth of the corresponding pinion, taking a given angular amplitude θ_2, of vibration of the gear wheel.

n = frequency of gear wheel vibration.
R = gear wheel radius.
r = pinion radius.
T = torque at pinion.
I = inertia (moment) of turbine rotor.

The vibratory motion of the gear wheel may be expressed thus as a function of the time t :—

$$\text{Angular displacement } \theta = \theta_2 \cos 2\pi n t.$$

Differentiating twice, we have:

$$\text{Angular acceleration } \ddot{\theta} = -(2\pi n)^2\theta$$

Whence the maximum value of the linear acceleration of the gear wheel teeth is

$$(2\pi n)^2 R\,\theta_2$$

Taking a single reduction gear first, if the load on the pinion teeth were removed for a short interval as a result of the vibration of the gear wheel then the power would be absorbed by the inertia of the rotor, which would thus accelerate, and the linear acceleration of the pinion teeth would be:

$$r \times \frac{T}{I}$$

The required condition, namely, that the gear wheel teeth cannot get away from the pinion teeth, is that this acceleration shall be greater than the maximum acceleration of the gear wheel teeth, or:

$$\frac{r}{R} \times \frac{T}{I} > (2\pi n)^2\theta_2$$

To take now the case of double reduction gear, let R_1, r_1 be the radii of the first and R_2, r_2 the radii of the second reduction gears, and let J be the inertia of the intermediate gear element. If a is the acceleration of the intermediate gear when it is free to accelerate along with the turbine rotor, then the corresponding acceleration of the rotor is

$$\frac{R_1}{r_1} a$$

Therefore

$$I \times \frac{R_1}{r_1} \cdot a + J a = T$$

or

$$a = \frac{T}{\frac{R_1}{r_1} \cdot I + J}$$

and we have as the condition that the gear wheel teeth cannot come out of contact—

$$\frac{r_2}{R_2} \times \frac{T}{\frac{R_1}{r_1} \cdot I + J} > (2\pi n)^2\theta_2$$

Relative to a single reduction gear, the double reduction gear is thus, on this basis, unfavorably conditioned. Its defect is, principally, the low potential acceleration of the second reduction pinion teeth. It is probable, again, that the conditions determining the factors n and θ_2 are favorable to the single reduction gear.

It appears impossible to treat mathematically the motions of the different elements of a geared turbine installation when the propeller shafting undergoes torsional vibration, but it is quite possible to obtain comparative estimates of the outside limits to the amplitude of vibration in the gearing. The method proposed rests on the assumption that vibrations arriving at the gearing are absorbed solely by the main gear wheel which is keyed on the propeller shafting. The inertia of the other parts of the machinery being thus neglected, the estimated gear wheel amplitude will certainly be too great, but by treating different designs on the same basis a useful comparison can be made.

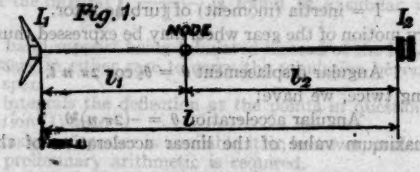


Fig. 1.

With the assumption stated, we have the simple vibrating system, of which Fig. 1 is a diagram, consisting of the shafting with the propeller on one end and the gear wheel on the other. Let

n = natural frequency of vibration (per sec.)
d = diam. of shafting, inches.
l = length of shafting, inches.
N = modulus of rigidity, lbs./sq. inch.
I_1 = inertia of propeller, lb.-ins.^{2}.
I_2 = inertia of gear wheel, lb.-ins.^{2}.
g = gravity acceleration, ins./sec.^{2}.

The node divides the length of the shafting into two segments l_1, l_2, which always twist in opposite angular directions, and we have then:—

$$2\pi n = \sqrt{\frac{N}{l_1} \times \frac{\pi d^4}{32} \times \frac{g}{I_1}} = \sqrt{\frac{N}{l_2} \times \frac{\pi d^4}{32} \times \frac{g}{I_2}}$$

Evidently

$$I_1 l_1 = I_2 l_2$$

(We have neglected the inertia of the shafting, for which it is not worth while making any correction.) Let

θ_1 = angular amplitude of propeller
θ_2 = angular amplitude of gear wheel.

For equilibrium at the node, the torsional strain must be the same in both segments of the shafting:

$$\frac{\theta_2}{l_2} = \frac{\theta_1}{l_1}$$

Therefore:

$$\theta_2 = \frac{I_1}{I_2} \times \theta_1$$

This brings us to the question of the propeller's vibration, of which θ_1 is the amplitude. Suppose the propeller is subjected to a single isolated shock, as by a sudden change in the resisting torque to a higher or a lower value, then the propeller responds with its natural frequency n, the vibrations being gradually damped away by friction; probably such vibrations are always small in amplitude. But when the disturbing force on the propeller is periodic as is actually the case for all ships, the resulting amplitude of the propeller may be very great. Let the variation in the resisting torque T (in.-lb.) be represented by a harmonically-varying disturbance of amplitude T, frequency m. Thus T is the average value and T cos $2\pi m t$ is the disturbance of the torque T + T cos $2\pi m t$. This periodic force is applied to the propeller, which has a natural frequency n. The solution of the equation of motion brings out very clearly the nature of the resulting vibration.* It may be written:

$$\theta = T \times \frac{g}{I_1} \times \frac{\cos 2\pi m t}{(2\pi n)^2 - (2\pi m)^2}$$

The frequency is the same as that of the force, and the amplitude is:

$$\theta_1 = T \times \frac{g}{I_1} \times \frac{1}{(2\pi n)^2 - (2\pi m)^2}$$

$$= \frac{T}{I_1} \times \frac{9.8}{n^2 - m^2}$$

We have neglected friction, of which the effect increases with the amplitude. If m and n are nearly equal, friction may prevent the amplitude getting so great as to cause destruction of the shafting.

With a four-bladed propeller

$$m = 4 \times \frac{\text{R. P. M.}}{60}$$

assuming that there is one radius in the wake stream where the thrust is a maximum and another where the thrust is a minimum.

From the expressions already given for n in terms of I_1, l_1 or I_2, l_2, we easily find:

$$n = \frac{1}{2\pi}\sqrt{\frac{N}{l} \times \frac{\pi d^4}{32} \times g\left(\frac{1}{I_1} + \frac{1}{I_2}\right)}$$

$$= 3{,}400 \times \frac{d^2}{\sqrt{l}} \times \sqrt{\frac{1}{I_1} + \frac{1}{I_2}}\text{; taking N}$$

$$= 12 \times 10^6 \text{ lbs./sq. in.}$$

We have, finally,

$$\theta_2 = \frac{I_1}{I_2} \times \theta_1 = \frac{T}{I_2} \times \frac{9.8}{n^2 - m^2}$$

This result indicates the benefit of a large and heavy gear wheel. We are not concerned with the sign of $n^2 - m^2$ which only determines the phase of the vibration relative to the phase of the torque variation. In most ships with propelling machinery of the type in question, m is greater than n. It will be seen that with this condition increase of propeller inertia exercises a small effect in reducing the overrunning tendency at the gearing; on the other hand, when m is less than n, an increase of propeller inertia has the opposite effect.

*Rayleigh's Theory of Sound, vol. i.

A simple relation may be written connecting torque variation and wake variation. Average torque:

$$T = \text{constant} \times \frac{\text{S. H. P}}{\text{R. P. M.}}$$

For any given propeller,

$$\frac{\text{S. H. P.}}{(\text{R. P. M.})^3} = \text{constant, approximately.}$$

Therefore:

$$T = \text{constant} \times (\text{R. P. M.})^2.$$

Again, for a given pitch, slip ratio, and speed of ship:

$$\text{R. P. M.} = \text{constant} \times \lambda$$

where

λ = average value of wake factor

$$= \frac{\text{speed of ship relative to wake}}{\text{speed of ship relative to still water.}}$$

hence

$$T = K\lambda^2$$

where K is a constant.

Now the torque variation of amplitude T is consequent on a wake variation of amplitude μ, say. So we can write

$$T + T\cos 2\pi m t = K(\lambda + \mu \cos 2\pi m t)^2$$

$$= K(\lambda^2 + 2\lambda\mu \cos 2\pi m t) \text{ approximately}$$

whence:

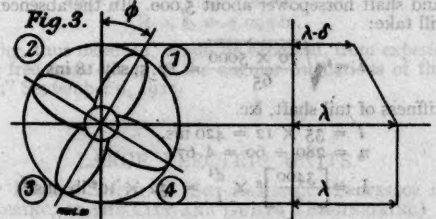
$$T = K \times 2\lambda\mu$$

$$= \frac{2\mu T}{\lambda}$$

μ is the amplitude of wake variation affecting the propeller as a whole. It must be much less than the amplitude of wake variation experienced by any one blade, because when one blade is in the region of minimum wake another blade will be in the region of maximum wake; in fact, if the wake be supposed to vary according to a simple mathematical law, there may be no wake variation experienced by the propeller as a whole. Regarding the way the wake actually varies over the section of the propeller stream few experimental data are available, but it is not obvious in any case how μ could be found from such data.

Let us start with the simple assumption that the wake factor varies uniformly from $\lambda - \delta$ at the top of the propeller to $\lambda + \delta$ near the keel. The sum of the wake variations for four blades is (see Fig. 2):

$$\delta\cos\phi + \delta\cos\left(\phi + \frac{\pi}{2}\right) + \delta\cos(\phi + \pi) + \delta\cos\left(\phi + \frac{3\pi}{2}\right) = 0.$$



Let the wake factor be supposed constant over the bottom half of the propeller disc, and to vary uniformly from the center to the top (Fig. 3), then the wake factor for each blade is as follows:

Blade.	Wake Factor.
(1)	$\lambda - \delta \cos \phi$
(2)	$\lambda - \delta \cos \left(\phi - \frac{\pi}{2} \right)$
(3)	λ
(4)	λ

Hence, wake factor for the whole propeller =

$$\lambda - \frac{\delta}{4}\left[\cos \phi + \cos \left(\phi - \frac{\pi}{2} \right) \right]$$

$$= \lambda - \frac{\delta}{4}\left[\cos \phi + \sin \phi \right]$$

We are considering one-quarter revolution. The greatest value is $\lambda - \frac{\delta}{4}$ and the minimum is $\lambda - \frac{1.4\,\delta}{4}$. Hence, let the expression be replaced by the harmonic function:

$$\lambda - \frac{1.2\,\delta}{4} + \frac{0.2\,\delta}{4} \cos 2\pi m t$$

Thus,

$$\mu = \frac{0.2\,\delta}{4} = 0.05\,\delta.$$

Suppose $\delta = 0.4$, then $\mu = 0.02$.

The foregoing is no doubt rough mathematics, but serves to show that a large variation in the wake stream has but a small effect in relation to torque variation. In an interesting paper by Messrs. Walker and Cook, read before the Institution of Naval Architects this year, observations are recorded of the actual fluctuations of twist of a propeller shaft. The ship in question was an oil tank steamer, the *San Fernando*, in which trouble had been encountered as a result of torsional oscillations. It was found that the shafting vibrated with a natural frequency which corresponded to a nodal position at the gearing. This frequency was about 280 per minute. The revolutions, originally 65 to 70, were reduced to 61.3 by increasing the pitch of the propeller. The twist fluctuation was then observed by moving the torsionmeter scale box to different angular positions round the shaft, and was found to have an amplitude equal to about 33 per cent of the mean twist. The length of shafting was 35 feet, and shaft horsepower about 5,000. In the absence of more particulars, we will take:

$$d = \sqrt[3]{\frac{70 \times 5000}{65}} = 17.5, \text{ say } 18 \text{ ins.}$$

to allow for stiffness of tail shaft, &c.

$$l = 35 \times 12 = 420 \text{ ins.}$$
$$n = 280 \div 60 = 4.67.$$
$$I_1 = \left[\frac{3400}{n}\right]^2 \times \frac{d^4}{l} = 132 \times 10^6 \text{ lb.-ins.}^2$$

corresponding to a propeller about 22 feet diameter and 45,000 pounds weight.

$$T = 63{,}000 \times \frac{5{,}000}{61.3} = 5.14 \times 10^6 \text{ in.-lbs.}$$

$$\text{Twist} = \frac{5.14 \times 10^6}{12 \times 10^6} \times \frac{420 \times 32}{18^4 \times \pi} = 0.0174$$

$$\theta_1 = 0.33 \times 0.0174 = 0.00574$$
$$m = 4 \times 61.3 \div 60 = 4.09$$
$$T = I_1 \theta_1 \times (n^2 - m^2) \div 9.8$$
$$= 0.4 \times 10^6 \text{ in.-lbs.}$$

which is about 8 per cent of the average torque.

$$\mu = \frac{1}{2} \times \lambda \times \frac{T}{T} = 0.027 \text{ with } \lambda = 0.7$$

This is a full-lined ship, and the value of μ will be high. The provisional figure, 0.02, already arrived at, is therefore probably quite a close approximation for fine-lined or twin-screw ships.

The system of comparison here indicated has, in a number of cases, shown very consistent results. Linear amplitude of gear wheel teeth = $R \times \theta_2$, where R is main gear wheel radius. A gear with a low value of $R \times \theta_2$ is found to be silent and to run satisfactorily; a high value of $R \times \theta_2$ indicates a noisy and troublesome gear.

Let a case be proposed where shaft horsepower = 2,500, revolutions per minute = 80. Propeller diameter about 17 ft., giving $I_1 = 40 \times 10^6$ lb.-in^2. Shafting diameter d = say, 12¾ in., length l = 110 ft. × 12 = 1,320 in. Gear wheel radius R = 40 in., $I_2 = 18 \times 10^6$ lb.-in.^{2}:

$$m = 4 \times 80 \div 60 = 5.33$$

$$n = 3{,}400 \times \frac{12.75^2}{\sqrt{1320}} \times \sqrt{\frac{1}{40 \times 10^6} + \frac{1}{18 \times 10^6}} = 4.3$$

$$T = 63{,}000 \times \frac{2{,}500}{80} = 1.97 \times 10^6 \text{ in.-lbs.}$$

Take $\lambda = 0.7$, $\mu = 0.028$, then

$$2\frac{\mu}{\lambda} = 0.08$$

Hence $$T = 0.08\ T = 0.157 \times 10^6$$

$$\theta_2 = \frac{0.157 \times 10^6}{18 \times 10^6} \times \frac{9.8}{5.33^2 - 4.3^2} = 0.0086$$

and $R \times \theta_2 = 0.34$ in., which is too high a figure. Let the design be modified to reduce the value of $R \times \theta_2$. Increase the gear wheel radius to 50 inches, thus making I_2 = about 46×10^6 lb.-in.^{2}, and reducing n to 3.29. Then:

$$\theta_2 = \frac{0.157 \times 10^6}{46 \times 10^6} \times \frac{9.8}{5.33^2 - 3.29^2} = 0.0019$$

$$R \times \theta_2 = 0.095 \text{ in.}$$

and the gearing, thus modified, would be expected, from experience with this theory, to be free from effects of the angular oscillations of the propeller.—"Engineering," September 9, 1921.

LARGE ELECTRIC UNITS.*

By Stanley Parker Smith, D. Sc., Assistant Professor in Electrical Engineering, at the City and Guilds (Engineering) College.

Introducing Note on Prime Movers.—The development of prime movers has largely influenced the development of electric generators, and conversely. Transmission problems and the growing demand for electric energy have given a great impetus to alternating-current machinery.

Steam Turbines have now reached a high stage of development and can meet practically all needs as regards output and speed. Units of 25,000 kilowatts to 35,000 kilowatts are probably large enough to give good all-round economy without involving too much capital expenditure in spare plant and without causing too much disturbance in the event of failures. One of the most interesting developments of recent years has been the Ljungström steam turbine, in which the electric unit is split into halves. By making these halves rotate in opposite directions, the relative blade speed in the turbine is made equal to twice the actual running speed, resulting in a very small turbine, low steam consumption, and the possibility of much larger units at 3,000 revolutions per minute than with ordinary turbines, because each generator gives only half the output of the unit. The two stator windings are connected permanently in parallel and the two rotor windings in series. The set is started with the exciter circuit closed and the two alternators synchronize automatically at about half speed. The whole set can be mounted on the condenser.

A Ljungström set by the Brush Electrical Engineering Company is shown is Fig. 1.

Large Internal-Combustion Engines have been widely developed on the Continent. Gas engines are now installed in most iron and steel works to utilize blast-furnace gas. The exhaust gas from the engines can be further utilized by using it in gas-fired boilers, after which it can be used for heating the feed water—the extra power thus obtained from the steam turbines may amount to one-eighth or more of the total. Recent developments are illustrated by greater reliability, simplification of valve gears, use of compressed air for starting, &c. Flywheels may be necessary

* Paper read before Section G of the British Association at Edinburgh, September 13, 1921.

for reducing cyclic irregularities, but may be dispensed with in alternators driven by twin tandem engines. Regarding 2,000 horsepower per cylinder as the present limiting output, such a set could give 8,000 horsepower. An idea of the outputs and speeds obtainable with gas engines is obtained from the following figures for tandem engines:—

Cockerill: 1,000 horsepower at 115 revolutions per minute, to 3,600 horsepower at 94 revolutions per minute.

M.A.N., 715 horsepower at 167 revolutions per minute, to 3,500 horsepower at 94 revolutions per minute.

Twin-tandem (four-cylinder) engines give about twice the above powers. These figures may be compared with the following figures for Diesel engines with one to six cylinders:—

Sulzer: 20 horsepower at 260 revolutions per minute, to 4,000 horsepower at 125 revolutions per minute.

M.A.N.: 85 horsepower at 200 revolutions per minute, to 1,500 horsepower at 150 revolutions per minute.

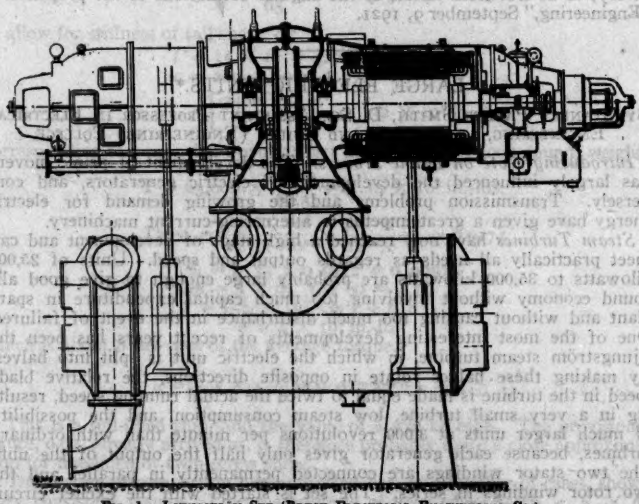


FIG. 1. LJUNGSTRÖM SET (BRUSH ELECTRICAL ENGINEERING CO.)

Water Turbines.—In no other direction has development been greater than in hydroelectric work. The design of both impulse and reaction turbines has been greatly improved as a result of experience. Heads of 3,000 feet and units of 20,000 horsepower are becoming quite common. So long as coal was relatively cheap and easily obtained, development was rather slow, but the coal famine of recent years proved sufficiently powerful to overcome the adverse financial conditions. In many countries definite policies are being carried out in connection with water-power schemes—*e.g.*, to secure continuity of supply, the Alpine stations in Italy are linked up with the Apennine stations; while in Switzerland, high-head stations with storage capacity are linked up with low-head, river stations working on constant load. In Germany, power is obtained at the weirs of canalized

rivers. Railways are being worked electrically by power obtained from hydraulic stations in Sweden, Switzerland, Austria and Italy. The greatest difficulty in the construction of alternators for this work is the need for safeguarding them against overspeeds, which vary from 80 per cent to well over 100 per cent according to the type of turbine, &c.

PRODUCTION OF ALTERNATING-CURRENT ENERGY.

1. Turbo-Alternators.

Rotors.—The difficulties of construction, now well understood, have increased with increasing outputs. Three thousand revolutions per minute has become almost the standard speed for 50-cycle machines, and outputs up to 20,000 k.v.a. per unit are now practicable at this speed. This entails peripheral speeds up to, or even beyond, 25,000 feet per minute. Despite the difficulties in obtaining large forgings weighing 20 tons to 25 tons in their finished condition, the solid rotor seems to be generally preferred to any form of built-up construction. As a result, dynamo makers both at home and abroad are insisting on more rigorous guarantees from steel makers, and a hole has generally to be bored or trepanned through the shaft to enable the interior of the forging to be examined and to provide test-pieces. The provision of efficient ventilation channels becomes very difficult in large sizes, and there is a marked tendency to rely exclusively on surface cooling, safety being ensured by non-combustible insulation. Water-cooled rotors are built, but such cooling may become superfluous if the simpler method proves satisfactory.

Despite mechanical weaknesses, the built-up rotor enables all parts to be thoroughly examined during construction and simplifies the provision of ventilation ducts. For large four-pole rotors, weighing 35 tons to 45 tons, some form of built-up construction appears to be necessary. Among the special constructions may be mentioned the parallel-slot rotor, and the salient-pole rotor. An interesting example of the latter is the arrangement by Ganz, in which grub screws are made to secure a solid pole-shoe and a sleeve which carries the winding (see Fig. 2). In large machines, particularly of the four-pole type, the rotor end-balls are often troublesome to design.

Illustrations of various types of rotors are given in the accompanying figures.

Stators.—Ventilation problems are here accentuated by the greater losses, and particular care has to be taken to keep down additional losses, especially in the embedded copper. For large sizes, neither simple axial nor simple radial ventilation is alone sufficient, but the air must be made to flow in some preordained manner, by some suitable arrangement. The manufacturer's difficulties in this country are often enhanced by having to build the stator on site. Water cooling is also being attempted to assist stator cooling. Station engineers are not yet wholly satisfied with the methods adopted for cleansing the cooling air—the method of circulating the cooling air through a refrigerator is being tried in several places. In the Schneider water-cooled stator, the water is first passed through the refrigerator used for cooling the circulating air. After absorbing heat from the air and from the machine, the water is then used to heat the feed water.

The end windings call for special consideration, and the state of the art is now such that turbo-alternators can safely withstand sudden short-circuit conditions, though in low-frequency systems an external reactance coil is sometimes considered a desirable additional safeguard. Both

single- and double-layer stator windings are widely used, and with the latter arrangement, the coil-span is often adjusted to suit design conditions. Automatic voltage regulation is becoming universal, the popular methods being those which act on the exciting current of the exciter.

2. Water Wheel Alternators.

The main difficulty in all water-wheel-driven generators arises from the runaway speed, which may be considerably over 100 per cent above the normal speed. Five diagrams have been prepared from the information supplied by the several firms, and show the great variety of methods used for securing the poles and windings. These are given in Figs. 2 to 21. As in turbo-alternators, the stator winding is usually well braced to prevent accidents in the case of sudden short circuit.

Most modern types of water-wheel alternators are entirely enclosed to reduce windage loss and noise, and to protect the machine against eventual leaks. Large flywheel effect is generally important, and the water-turbine manufacturer is generally anxious to have the necessary flywheel effect embodied in the rotor of the alternator, so as to dispense with a separate flywheel—a condition not always easily fulfilled by the dynamo builder.

A few days ago, the author saw two further interesting modes of construction—a robust construction by B.B.C. in which the pole cores were screwed into sleeves protruding from the yoke, and an Oerlikon construction in which the pole laminations were inserted between solid projections from the yoke and held by two large pins.

3. Slow-Speed Alternators.

These machines have become important again (after their suppression by the steam turbine) owing to the development of the internal-combustion engine and the adoption of reduction gearing. The large flywheel effect needed to obtain steady running with gas and oil engines often entails an "outer-pole" type of construction.

An illustration of a gas engine built by the Société Anonyme John Cockerell driving a 2,200-k.v.a., 2,000-volt generator, with 64 poles running at 94 revolutions per minute built by the Charleroi Co. is shown in Fig. 22.

PRODUCTION OF CONTINUOUS-CURRENT ENERGY.

The following alternative methods for producing continuous-current energy will be considered:—

1. Direct-driven, slow-speed generators.
2. Slow-speed generators with reduction gearing.
3. Rotary and motor converters.
4. Mercury rectifiers.

The continuous-current turbo-generator may be regarded as dead. After being developed successfully at enormous cost, it has been ousted by more suitable apparatus.

The direct-driven continuous-current generator owes its extension of life largely to the development of the internal-combustion engine.

Probably the sternest competitors at the moment are the slow-speed generators driven by steam turbines through reduction gearing, and the combination of turbo-alternators with rotary converters. For large powers, the latter are probably cheaper and the more economical; but for small powers, the geared generator appears to many to be the ideal arrangement. Gearing has now been brought to a high pitch of efficiency and

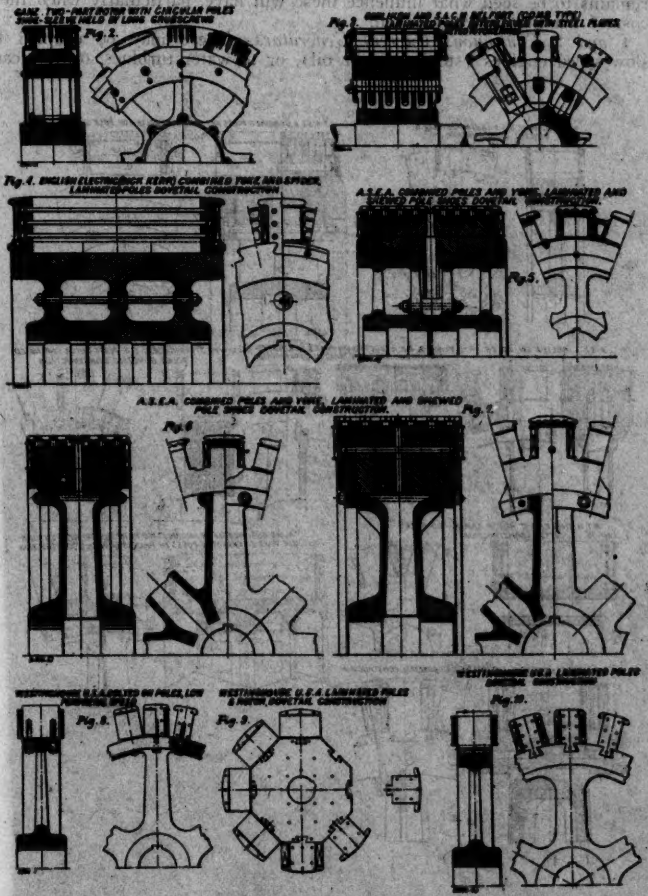
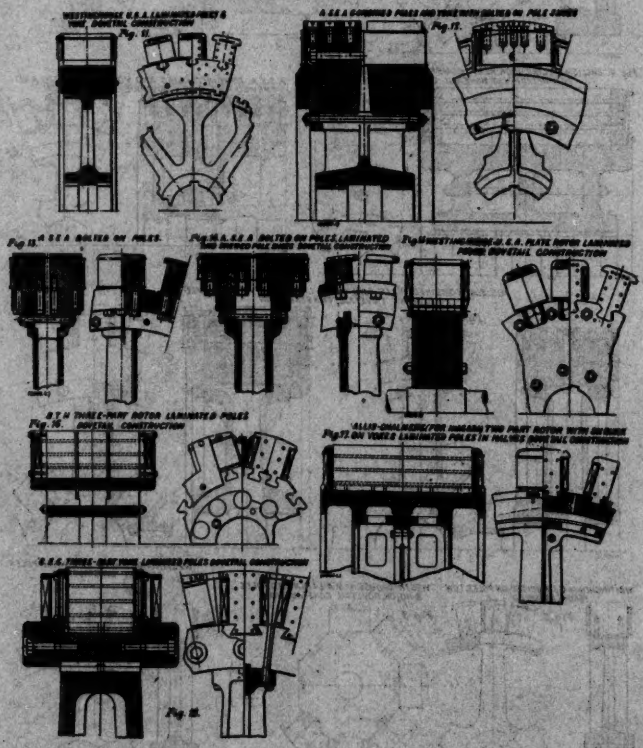


Fig. 2.
Fig. 3.
Fig. 4. ENGLISH ELECTRIC (BIRCH, KEMP) COMBINED YOKE AND SPIDER, LAMINATED POLES DOVETAIL CONSTRUCTION.
A.S.E.A. COMBINED POLES AND YOKE, LAMINATED AND SHEWED POLE SHOES DOVETAIL CONSTRUCTION.
Fig. 5.
A.S.E.A. COMBINED POLES AND YOKE, LAMINATED AND SHEWED POLE SHOES DOVETAIL CONSTRUCTION.
Fig. 6
Fig. 7.
Fig. 8.
WESTINGHOUSE U.S.A. LAMINATED POLES 8 POLES DOVETAIL CONSTRUCTION
Fig. 9
Fig. 10.

reliability, and permits the turbine to be run at a speed consistent with low steam consumption, and a generator which is economical both in cost and working.

Mercury rectifiers have only lately come on to the market, and it remains to be seen what influence these will have, particularly when their cost of manufacture is reduced.

1 *and* 2. *Continuous Current Generators.*—These may be coupled to slow-speed engines (steam, gas or oil), or to water turbines, or to steam



turbines through reduction gearing. Illustrations of continuous-current generators are given in Figs. 23 to 25. The first of these shows a double commutator armature of 1,000 kilowatt, running at 94 revolutions per minute, constructed by the Société Alsacienne de Constructions Mécaniques à Belfort. It is used for three-wire distribution. Fig. 24 shows a water turbine-driven 3,000-kilowatt generator for 300 volts. It generates 10,000

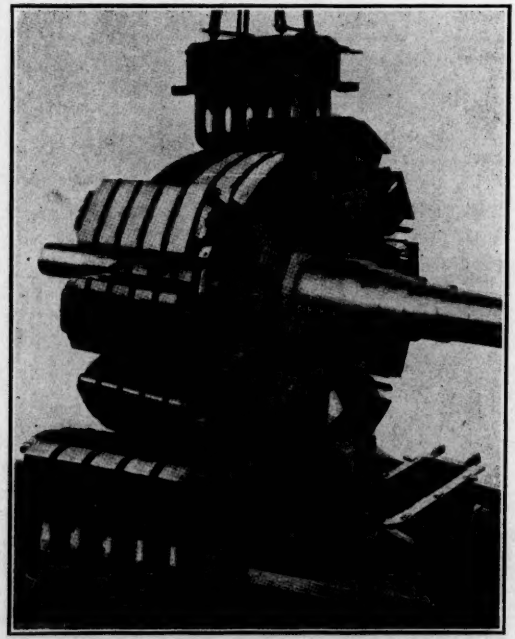


FIG. 19.—WATERWHEEL ROTOR; COMBINED CONSTRUCTION (OERLIKON).

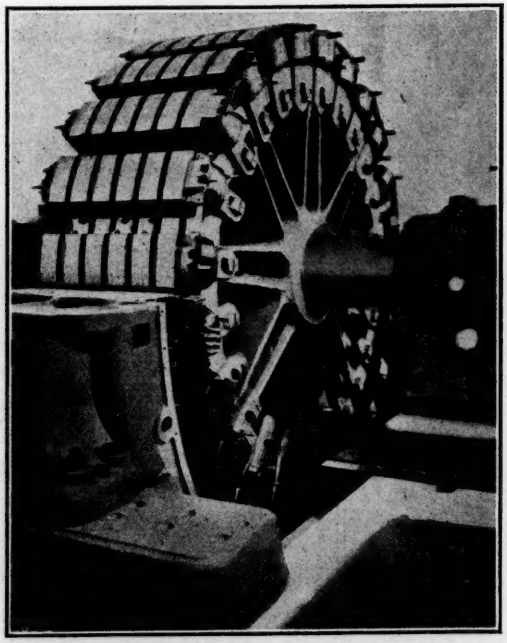


FIG. 20.—WATERWHEEL ROTOR; COMBINED CONSTRUCTION (S.A.C.M.).

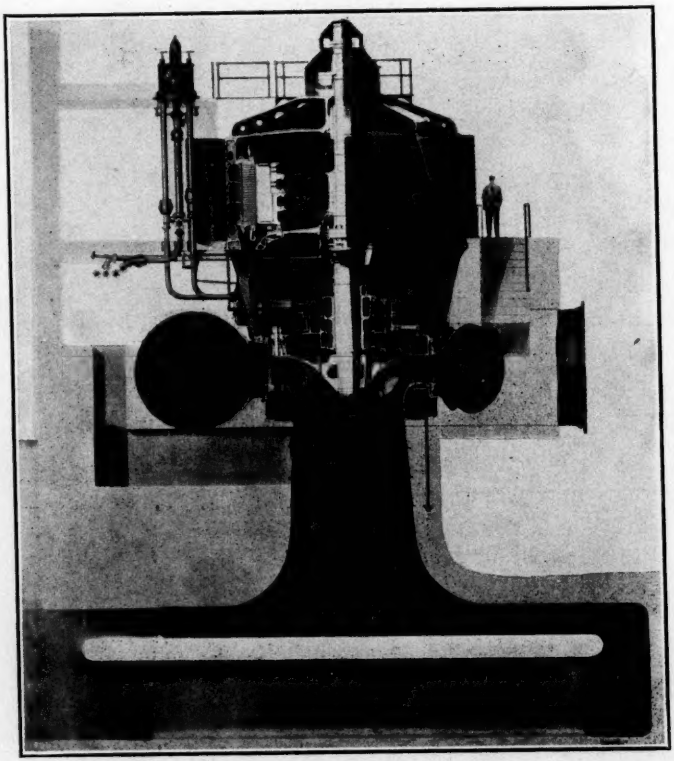


FIG. 21.—WATERWHEEL ALTERNATOR FOR NIAGARA 32,500 K.V.A. (ALLIS-CHALMERS CO.).

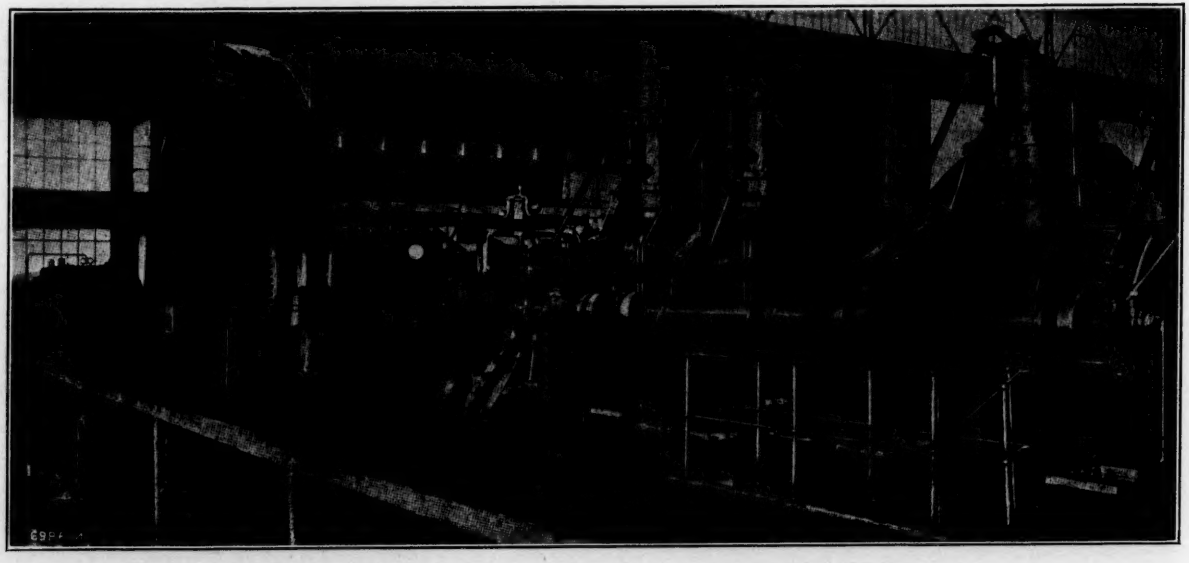


FIG. 22.—GAS ENGINE DRIVEN ALTERNATOR (J. COCKERILL AND A.C.E.C.).

FIG. 23.—DOUBLE COMMUTATOR ARMATURE (S.A.C.M.).

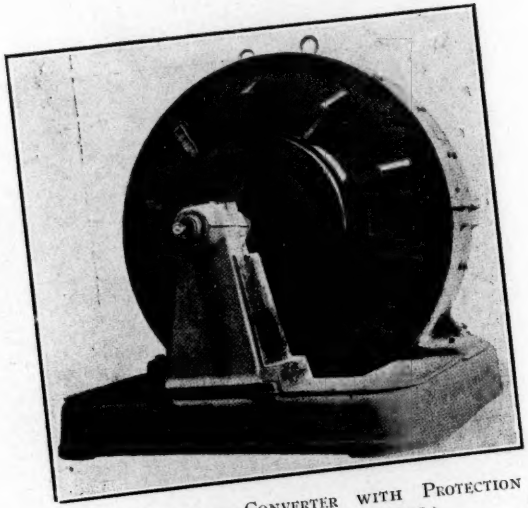


FIG. 26.—ROTARY CONVERTER WITH PROTECTION AGAINST FLASH-OVER (B.T.H.).

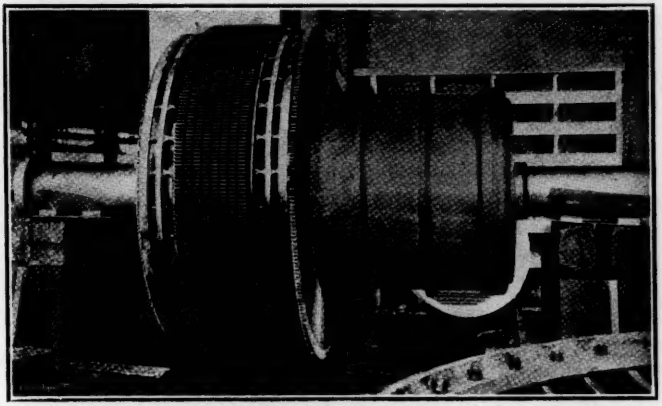


FIG. 24.—LARGE COMMUTATOR WITH SHRINK RINGS FOR WATER-TURBINE DRIVE (ALLIS-CHALMERS).

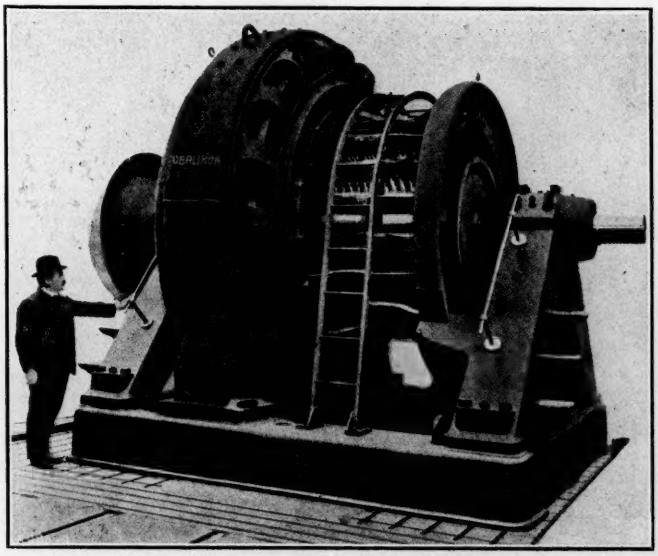


FIG. 25.—3,000-KW. 300-R. P. M. C. C. GENERATOR DRIVEN BY WATER TURBINE (OERLIKON).

ampères and runs at 300 revolutions per minute, and was built by the Allis-Chalmers Company. The third example, shown in Fig. 25, is a 3,000-kilowatt, 375-volt, 300-revolutions-per-minute set, constructed by the Oerlikon Company.

3. *Rotary and Motor Converters.*—Large converters started from the alternating-current side have usually a starting motor with two poles less than the converter. The starting motor is of the induction type and is commonly supplied with a solid cast-iron rotor for cheapness. In order to synchronize correctly automatically, it is usual to connect an impedance in series with the armature when applying the alternating pressure. One common way of doing this is to utilize the stator winding of the starting motor, by opening the neutral point and joining the phases in series with the slip-rings. Another method is to insert a choking coil in series with the slip rings. By these means the current is limited to a value which is insufficient to reverse the residual magnetism in the converter, which thus synchronizes correctly. By increasing the induction motor field when the set approaches synchronous speed, Messrs. Mather and Platt have found that the converter will synchronize when the motor has as many poles as the converter—in this way, surging and sparking during the transition period are prevented.

Damping windings are universally used to prevent hunting, but experience has shown that the bars must be spaced so as not to produce fluctuations in the continuous pressure, because of the danger of telephonic interference. Various methods are in use for obtaining voltage regulation, including reactance control, booster control, induction-regulator control, and the much rarer methods such as the split-pole converter.

The difficulties connected with the design of rotary converters for 50 cycles and 60 cycles have been largely overcome, but with present limits it is difficult to build rotary converters for high pressures. Thus at 50 cycles, it is not easy to build a rotary converter for more than 1,000 volts. This matter is important for electric traction where it is desired to obtain 1,500 volts at 50 cycles. In this case it may become necessary to use either motor generators, two rotary converters in series, motor converters, or mercury rectifiers; but mention should be made of the fact that the Oerlikon Co. recently built a successful rotary converter for 1,500 volts at 50 cycles.

The motor converter owes its development mainly to Messrs. Bruce Peebles. Owing to the fact that pressures up to 11,000 volts can be applied direct to the stator winding of the induction machine, a transformer becomes unnecessary with these sets in many cases. The development and use of converting plant appears to be more general in Great Britain and in the United States of America than on the Continent of Europe. The protection against flashing-over used by the British Thomson-Houston Company is illustrated in Fig. 26.

4. *Mercury Rectifiers.*—After much experimental work, these have now been successfully developed by Messrs. Brown, Boveri and Co., and are now on the market in three sizes, namely, 300 ampères, 600 ampères and 1,000 ampères, at voltages up to 1,500. Rectifiers are also being built by German firms. Owing to the short time this apparatus has been in general use, it is too early to make a proper comparison with other types of converting machinery. From all accounts, however, it would appear that a very useful alternative method of converting alternating into continuous-current energy has now been made available. Slides shown at the meeting illustrated the Brown, Boveri rectifier.—"Engineering," September 16, 1921.

THE WELDING OF STEEL.*

The Welding of Steel in Relation to the Occurrence of Pipe, Blow-Holes and Segregates of Ingots.

By H. Brearley, Member of Council (Sheffield).

It is well known that perfectly clean surfaces of malleable metals will stick together under pressure at temperatures much below those commonly used for welding. Hot steel can be welded together under pressure at low temperatures in an atmosphere of hydrogen or other inert gas. Turnings sometimes stick to the cutting edge of tools which are doing heavy work, and it is on record that a cutting tool has been found welded to a heavy forging which was being machined.

Steel oxidizes at comparatively low temperatures. The temper colors visible on bright steel heated up to 200 degrees C. and beyond are forms of iron oxide which, as the temperature is raised, pass gradually into the more or less adherent oxide of measurable thickness which we call scale. The scale thus formed is a brittle substance whether hot or cold; it is not malleable or plastic, and cannot be welded together. At very high temperatures the scale melts and could be expressed in the fluid condition from between two metallic surfaces, but the temperature at which it remains sufficiently fluid is near the melting point of steel. At such high temperatures the fluid oxide scale reacts with the carbon of the plastic steel, liberating carbon monoxide or dioxide, which forms blisters. It is, therefore, not practicable to make a respectable weld without using some means of lowering the melting temperature of the scale, and also modifying its oxidizing effect of the hot steel.

In order to flux the scale which forms on heating iron and steel, the blacksmith uses sand. The sand-scale slag thus formed should be such as is fusible at temperatures some hundreds of degrees below the melting point of either the scale or the sand. Its fusibility varies with the respective proportions of scale and sand used. Too much of either constituent produces a less fusible flux. Since the amount of one of the constituents—the scale—cannot be determined, the most effective quantity of sand to use can be selected only by skilled judgment. It depends on the amount of scale needing to be fluxed, and the area to be covered by the fusible sheath which cleans the surfaces and shields them from further oxidation.

Being produced under empirical conditions which are so difficult to control, there is no wonder that welds apparently similar are really of varying quality. The best of them are never perfect, though statements to the contrary are frequently made. A weld may appear under certain test conditions to be stronger than the adjacent part, for two reasons. First, because the adjacent parts have been badly overheated and not hammered; and second, because the welded part has been superficially carburized by the coke fire and to a corresponding extent that part has a higher tensile strength. Both these conditions are due to negligence, and hence, judging from the comparative behavior of welded and adjacent parts under the stress of a tensile test, we arrive at this amazing paradox—the less skilfully the welding is done the better the weld may appear to be.

The tensile test is not a very satisfactory means of recording the weldability of a steel, or the efficiency of a welding process. Although all welds are more or less defective the position of the defect may be unknown, and consequently when a tensile test-piece has been machined it is a question of pure chance whether a particular kind of defect is in the core of the test-

* Abstract of a paper read at the meeting of the Iron and Steel Institute on Thursday, May 5.

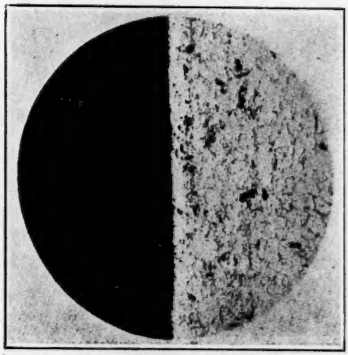


FIG. 1.—SECTION OF ROLL-WELDED PLANE IRON.

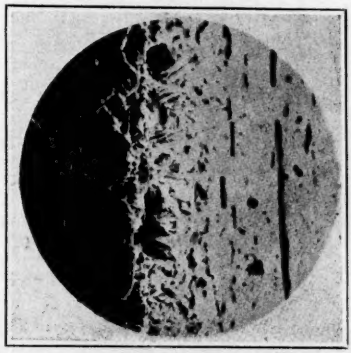


FIG. 2.—SECTION OF HAND-WELDED PLANE IRON.

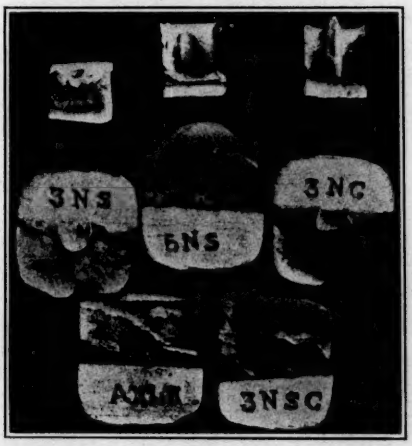


FIG. 3.—FRACTURE OF TOUGHENED BARS SHOWING CORES.

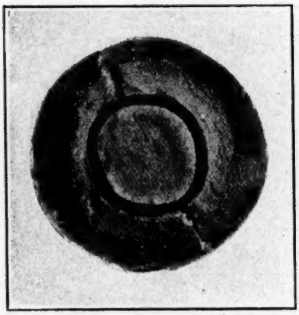


FIG. 4.—FRACTURED TENSILE PIECE SHOWING CORE.

FIG. 5.—FRACTURED TENSILE PIECE NO CORE VISIBLE.

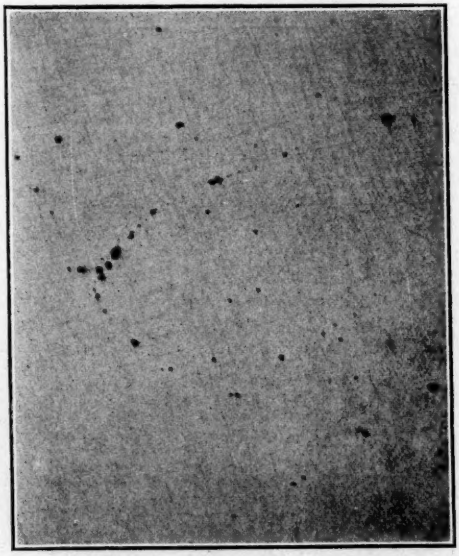


FIG. 6.—UNETCHED MICRO OF 3 PER CENT NICKEL STEEL "B." × 50.

FIG. 7.—ETCHED MICRO OF CARBON STEEL "A." × 250.

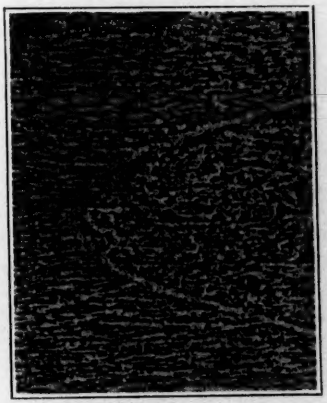


FIG. 8.—ETCHED MICRO OF 3 PER CENT NICKEL STEEL "B." × 50.

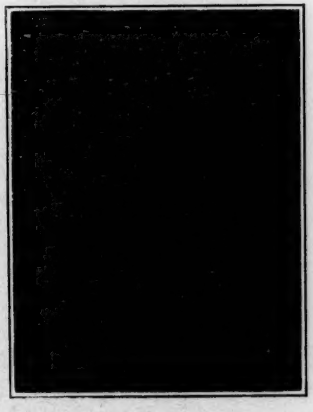


FIG. 9.—ETCHED MICRO OF 3 PER CENT NICKEL-CHROMIUM STEEL "D." × 50.

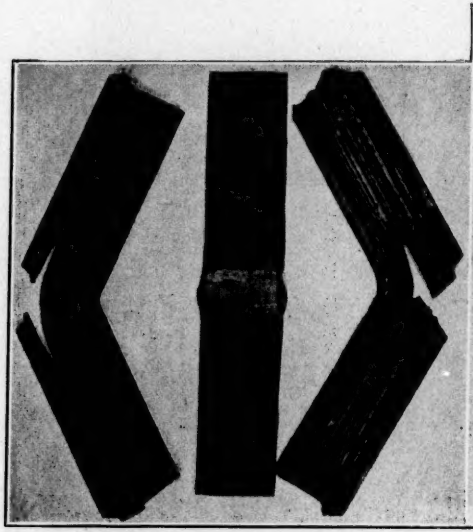


FIG. 10.—BROKEN IMPACT PIECES FROM CORED BARS.

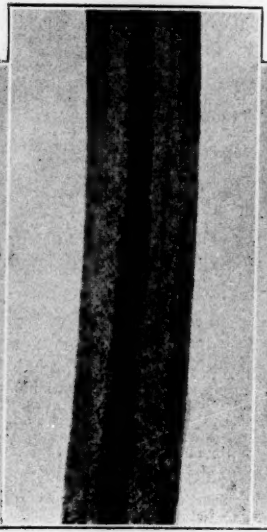


FIG. 11.—SEGREGATED BOILER PLATE.

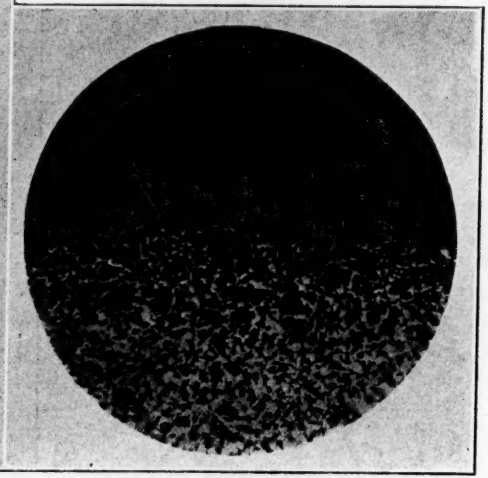


FIG. 12.—MICRO OF MACHINE KNIFE × 100.

piece or on its surface. In the former position the defect may not generally influence the test figures; in the latter position the test-piece breaks either before, or soon after, the maximum stress has been reached, and consequently the ductility figures are very poor. In the same way the tests made on gun-tube forgings, taken transversely, give very erratic results, due not to variations in the suitability of the material for its intended purpose, but to the occurrence of slag streaks haphazard, now in a favorable position, now in an unfavorable position in the test-piece.

Dr. Stead has stated that "the skilful microscopist can, with absolute certainty, determine whether or not surfaces are welded together, a matter of exceeding importance when considering whether cavities in steel ingots can, or cannot, be welded up." It is quite true that certain differences in welds become clear when sections are examined microscopically. The roll-welded plane iron in Fig. 1 is obviously not so good a weld as the hand-welded plane iron in Fig. 2. It is, however, merely a question of degree. The welded surfaces of Fig. 2 can be torn apart as completely as those of Fig. 1, but it would require a greater effort—neither of them are perfect welds.

How near to perfection a weld can be made is a question of great importance to steelmakers. Under rolling or forging conditions a steel may be unweldable and yet an ingot may be forged into sound bars. If, however, such an ingot be cracked, as more are than we like to admit, then it is not forgable, but will crumble under the hammer. This explains why of certain alloy steel ingots having the same composition some forge well, and others not at all, or very badly.

Ingots are never quite free from shrinkage cavities, contraction cavities, cracks, or blowholes. It has been proposed that ingots should be made with a volume of deep-seated blow-holes sufficient to neutralize piping and thus avoid the necessity of scrapping a discard head. As this method of avoiding pipe has been in use for many years it is fair to assume that such ingots produce billets, bars, and sheets, which are commercially satisfactory for certain purposes. Dr. Stead and others conclude from microscopic evidence "that the welding of blowhole cavities as a rule is complete and perfect." Their experimental evidence is obtained by boring a hole into an ingot or billet, fitting a plug of the same material, sealing the joint with a blowpipe, raising to welding heat and hammering or rolling into a bar. If the bar, when nicked and broken across the welded faces, show no signs of the weld, and the microscopic evidence be satisfactory, it is assumed that the weld has been perfectly made.

If by perfection we mean a welded joining which is in all respects as good as any unwelded part of the same bar, then it may be said that the evidence is insufficient to support the conclusion even so far as the limited experimental conditions are concerned. The conclusions are a still less satisfactory answer to the question whether blowholes in ingots weld up, because a drilled hole is smooth and clean, whereas a shrinkage cavity or blowhole may be uneven, lined with segregates, coated with alumina, or otherwise less favorably disposed to weld completely.

During the great war, trouble was caused by the occurrence in aero-engine crankshafts of small cracks whose origin was camouflaged by calling them "hair-lines"; just as, ten or twenty years ago, slag streaks were called "sand cracks" and many other names except slag. The so-called hair-lines arose mainly from ingot defects, notably those occurring in and about the axis of the ingot. These axial defects are the remnants of shrinkage taking place in the nearly solidified ingot. The small volume of impure metal last fluid moves downwards as best it may between the loosely adhering free crystals, filling up shrinkage and construction cavities as long as it is hot enough, or impure enough, to keep fluid. It is finally frozen in position, lying like an inverted cone about the axis of the ingot.

It is called a "V" segregate because it can be detected by a sulphide print but the harm lies in the small cavities themselves rather than in the sulphide segregate found in, or near, the cavities.

The "V" cavities are greater when an ingot is cast hot than they are when the same kind of ingot is cast cold. The remedy is to cast cold in a mould tapered as much as is practicable. The remedy, however, does not get rid entirely of the cavities, and one must reckon that the remnants of them may appear either as split centers, if the forging be carelessly done, or as small seams or "hair-lines," however skilfully the forging operation may be carried out.

It has been suggested that minute defects inherent in commercially sound ingots, and small blowholes which may occur accidentally, do not weld up in nickel-chromium steels, because steel containing chromium will not weld. And it has been said that if aero-crankshafts, and other important structural components, were made from nickel steels or, better still, from carbon steels, the greater weldability of these steels would remove all difficulties. When this idea is tested under conditions more favorable than those occurring in practice, *i.e.*, by drilling holes along the axis of a bloom, inserting a machined bar of the same material, and after hermetically sealing the ends, rolling the bloom into a small bar, which is then nicked and broken, we arrive at very curious results. Some steels tested in this way were ordinary mild steels containing 0.15 per cent and 0.35 per cent carbon, a 3 per cent nickel steel, a 5 per cent nickel case-hardening steel, and a nickel-chromium steel, such as was used for aero-engine crankshafts. On etched transverse sections of the bars the weld lines were either not visible or slight indications only were discovered after much patient searching. When the bars were nicked and broken across the welded surfaces, all the steels, judging from the fractures, appeared to have welded perfectly, except the 0.15 per cent carbon steel and the 5 per cent nickel case-hardening steel. Unless, therefore, we are prepared to believe that tough mild steels weld less perfectly than nickel-chromium steel, we must question the reliability of both these methods of measuring the efficiency of a weld.

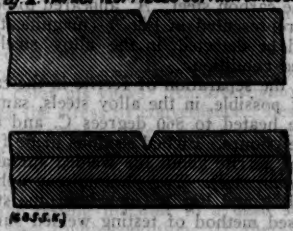
On hardening and reheating the entire series of steels so as to put all of them into a tough condition having Izod figures of over 50 foot-pounds, we find that not one of them withstands the nicked bar test; without exception they pull apart at the welded surfaces (Fig. 3). These results explain clearly why transverse fractures of nickel steel appear to be more reedy than similar fractures made in ordinary carbon steel, *i.e.*, the nickel steel is tougher and therefore pulls open on the faces of welded blowholes, or slag streaks, more than in the less tough carbon steels; the steel itself is not more reedy in the sense that it contains more slag streaks, or less perfectly-welded blowholes.

It is obviously incorrect to say that steel containing chromium cannot be welded under such conditions as apply to the welding of cavities in ingots. If, however, we are seeking some means of expressing the extent to which such steels weld, as compared with ordinary carbon steel, we should endeavor to state comparative results in concrete figures. No attempt appears to have been made to do this. The method to be described is submitted for consideration in the hope that it may be improved, or a better method may be evolved. Some method, at any rate, is desirable if we are to explain the known difficulties in forging special steel, and understand the effects of blowholes and segregates on the properties of steels containing them.

The steels to be examined are prepared as 3½-inch billets. A hole 1 inch in diameter is drilled down the axis of the billet, and a turned bar is driven

into it and fused over at each end. The cored billet is then rolled into a bar 4½ inches by ½ inch. Impact test-pieces cut from the side and center of this flat bar may be represented diagrammatically as in Fig. A. The test-piece cut from the side will be broken by a force recorded in foot-pounds which represents the effort required to start a crack at the root of the sharp notch and extend the crack across the specimen. The test-piece cut from the center of the bar is broken in the same way, but it will require a somewhat greater force to extend the crack across the specimen, depending on the perfection of the weld between the core and inside of the

Fig. A. IMPACT TEST PIECES CUT FROM CORED BARS



hollow billet. If no welding has occurred the crack will come to a dead stop when it reaches the core or leaves it, and a new crack would need to be started in each case. This, as we know, absorbs a great deal more energy than is required to propagate a crack which has been already started.

Very hard steels have low impact values because the material in and about the notch is not able to distort, and such steels, even in the form of cored bars, break readily across whether the weld is a good one or not. Conversely, steels having low impact values, whether they are very hard or not, fracture readily across welded cores, and this explains the insufficiency of the nicked bar test as an indication of good or bad welding. To pull a weld apart, if it can be done, the material should be put into a tough condition. The proposal is to measure the perfection of a weld in terms of the degree of toughness which must be induced in the steel before the weld can be pulled apart.

A series of specimens, numbered A to D, cut respectively from the edge and center of the flat bar, prepared as described, are hardened and tempered in stages of 100 degrees C. up to say 700 degrees C. These are broken on a standard Izod machine, and the impact value of the edge specimen whose corresponding piece cut from the center is split along the weld is taken as the welding figure.

From the results obtained the comparative welding figure selected would be: Steel "A" = 28 or rather less; steel "B" = 20; steel "C" = 18; steel "D" = 14 or rather less. That is to say, these figures represent in Izod foot-pounds the degree of toughness which must be induced in the steels by hardening and tempering in order to pull the welded surfaces apart into a visible gap.

If instead of rolling into flat bars the cored billets are rolled into round, or gothic, or square bars, and the tensile pieces are prepared from the bars which have been heat-treated to develop maximum toughness in them, the core will sometimes be visible in the broken tensile test-piece, and sometimes not visible. Of the above series of steels, each of which registered a

reduction of area of over 60 per cent, all exhibited a fracture like Fig. 4, except steel "C," whose fracture is represented in Fig. 5. This sample of 3 per cent nickel steel had, when tested, a maximum stress of 44 tons, an elongation of 29 per cent on 2 inches by 0.564 inch, and a reduction of area of 65.8 per cent.

For microscopic observation a transverse section could be cut from the flat bar on which, if the welded faces were well defined, the core would be represented by the outline of an ellipse. The pointed ends of such an elliptical outline could be traced with more or less certainty in all the specimens, but it was more marked in the alloy steels than in the carbon steels. In Fig. 6 the well-marked outline of steel "B" is represented. After etching, the weld in the carbon steel "A" appeared to be outlined by a band of ferrite such as is represented in Fig. 7 (magnification, 250 diameters). Nothing similar could be detected in the alloy steels in the rolled or hardened and tempered condition.

In order to increase the separation of ferrite in the carbon steel and promote its separation, if possible, in the alloy steels, samples of each of the four alloy steels were heated to 860 degrees C. and cooled down to 500 degrees C. in nine to ten hours. Etched sections made from these specimens now showed a band of ferrite about the welded surfaces of all the steels except the nickel-cromium steel. Two photomicrographs are reproduced in Fig. 8 (nickel steel "B") and Fig. 9 (nickel-chrome steel ("D").

How far the proposed method of testing welded surfaces, or determining the coefficient of welding, will be found reliable and useful must be shown by further experiment. It is clear, however, that all forms of plastic welding such as are used by the blacksmith, and such as operate in the rolling and forging of piped ingots, produce welded parts unlike other parts of the bar, inasmuch as the welded surfaces can be pulled open, and the free ferrite about them is quite abnormal.

The appearance of free ferrite about welded surfaces recalls the occurrence of the banded ferrite structure in mild steel plates and bars. The banded structure has been ascribed to micro-segregation of phosphorus. On the other hand it has been suggested that non-metallic impurities in steel in the form of slag streaks, &c., act as nuclei around which the ferrite gathers as it falls out of solution, just as sugar candy crystals form about a thread suspended in the liquor. Whilst the formation of free ferrite on the welded faces falls naturally into its place as a special instance of the effect of nuclei action, it is difficult to see how it can be related to phosphide segregation. Referring to Fig. 9, it may be suggested that the limited amount of free ferrite formed under the conditions of cooling was insufficient to form envelopes about the slag streaks and the welded faces. At any rate it should not be concluded that nickel-chromium steels behave exceptionally until lower carbon steels of the same type, or more slowly cooled steels in which free ferrite exists abundantly, have been observed.

The conditions, described in the earlier part of the paper, under which the coefficient of weldability is determined, are obviously empirical, and it would probably be necessary to adhere to given dimensions of billet, the inserted core, and the rolled flat bar in order to get consistent results. To observe the effect of deviations a 4-inch square billet was prepared with the usual 1-inch core, with a ½-inch core and with a ½-inch hole, the end only of which was plugged. In the rolled flat bar the welded surfaces would of course be a single one in the case of the ½-inch plugged hole, and double ones in the cases of the ½-inch and 1-inch cores. The welded surfaces would also lie variable distances below the bottom of the notch. The actual figures obtained both on softer and harder steels did not show great variations, and they were all in such directions as might be expected.

The effect may be summarized by saying that when the hardened and tempered pieces are sufficiently tough to cause the weld to split open, the impact value obtained from the welded pieces depends on the thickness of metal behind the split weld which has to be bent by the pendulum hammer. In Fig. 10 the right piece represents the ½-inch hole specimen and the left piece the 1-inch core specimen. It would clearly require more energy to bend the latter piece than the former after sufficient distortion had occurred to split the welded faces apart. Whilst, therefore, useful information may be obtained whatever size of billet and bar is used, it is advisable to adhere to fixed sizes if the results are required to be strictly comparative. The author prefers to insert a core into the drilled billet, as then two welded surfaces occur in the impact test-piece.

Although we may be convinced that no pipe or cavity in an ingot can be completely welded, we need be neither elated nor depressed by the fact. It is no part of an observer's business to allow personal prejudices or commercial prospects to color his views. An investigator's opinions ought not to be malleable under the stress of convenience. Besides nothing is wholly bad, and what may appear as a defect under one set of circumstances becomes an advantage under another. In this respect we may observe that the degree of imperfection of welds is made manifest by a rise in the impact figure. Were no welding to take place the impact test specimen used would bring the testing machine to a dead stop, although the material was not in a very tough condition. If, therefore, the material of a piped ingot, or a blown ingot, be stressed transversely to the direction in which it has been elongated by forging or rolling, such stresses carried to the point of fracturing will have more work to do than, under similar circumstances, would be required to fracture such material made from a sound ingot. The plate of a laminated spring is one of the best examples of the truth of this deduction, and as a matter of experiment it is found that a cambered spring plate made from the top of a piped ingot cannot be broken by scragging as easily as a similar plate made from the bottom end of the same ingot. In this sense blown ingots, or those rich in non-metallic impurities, make good spring plates, and it is mainly on this account that Bessemer steel is preferred by many spring makers to the more carefully melted and cast open-hearth steel. To some extent the same advantages apply to straight axles, and they are worth consideration even in relation to rolled tyres. It is very remarkable that the fabled virtue of all good wrought iron, viz., its fiber, should be regarded as a defect in steel. It should be valued of course like the grain in wood, according to its direction, in relation to the stresses put upon it.

That pipe, or blowpipes, or any other form of cavity in ingots is apt to be, and usually is, associated with segregated carbon, phosphorus, and sulphur, requires no special proof. If, therefore, the discussion be confined to the welding of ingot cavities it would be instructive to know how far the cohesion of the welded surfaces is influenced by carbon, phosphorus, and sulphur, respectively, and whether the influence of these elements present in the same specimen is additive or otherwise. A few statements on this subject have been made, but they must be regarded as opinions rather than as evidence. It has been said that "the welding up of blow-hole cavities in ordinary practice is, as a rule, complete and perfect," and also that "phosphorus concentration in blowholes facilitates welding, and iron relatively high in phosphorus is more readily welded than very pure material." We must remember, however, that this conclusion depends either on microscopic observation which may be illusory, or on the nicking and breaking method which may be misleading. Of the two methods of judging a weld it is claimed that "nicking and breaking was a more certain method than that of the microscope." As the nicking and breaking method

carried out on bars in the forged and rolled condition would lead us to believe that all hardening elements added to steel, including phosphorus, improve its welding properties, its results obviously need to be interpreted with caution.

It is generally stated, without any experimental warrant being adduced, that all segregates existing in ingot cavities decrease the weldability. It has been assumed that in the absence of segregates a perfect weld could be made, and as segregates and imperfect welds are often found together the presence of the segregate and the degree of imperfection are regarded as cause and effect. This conclusion is untenable now we know that perfect welds cannot be made even under the most favored conditions.

In 1912 Dr. Stead prepared two crucible ingots containing 0.020 per cent sulphur. One was blown and the other sound. Both were forged into round bars and parts of each bar were turned into annular rings. When drifts were forced into the rings those made from the blown steel broke with 50 per cent less extension than those made from sound steel. A sulphur print taken from the forged bars showed clearly the location of the blowholes because segregates had been pressed into them by such forces as operate in every solidifying ingot.

The weakness of the one set of rings as compared with the other was ascribed to the presence of sulphide segregates. Such a conclusion, however, is only justified if it can be shown that identical rings, made from blown steel containing no segregated sulphide, are practically as strong as rings made from sound steel. A direct experiment on these exact lines cannot be made, because it is impossible to produce commercial ingots containing blowholes without these holes being more or less lined with segregates. But as the blowholes could not weld completely under any circumstances it is likely that the rings were weaker in one case than the other, quite apart from any question of segregation.

To test this supposition a dozen small holes were drilled lengthwise to a depth of 8 inches into a 4-inch billet. Each hole was cleaned, dried, and filled with petrol vapor before being closed by plugs 1 inch long driven in with a sledge-hammer, and fused over with the oxy-acetylene blowpipe. The billet was rolled into a ⅞-inch round bar. From the sound and unsound parts of the round bars rings were turned. The rings were drifted, as in Dr. Stead's experiment, until they broke, with the results given in the next column.

Steel A. 865.

Carbon.	Silicon.	Manganese.	Sulphur	Phosphorus.
0.46	0.15	0.70	0.040	0.030

Condition of Steel.	Percentage Increase in Diameter of Drifted Rings made from "Unsound" Part.	"Sound" Part
As rolled	2.4	14.5
As rolled	3.4	12.6

Steel Y. 339.

Carbon.	Silicon.	Manganese.	Sulphur	Phosphorus.
0.15	0.23	0.92	0.060	0.018

Condition of Steel.	Percentage Increase in Diameter of Drifted Rings made from "Unsound" Part.	"Sound" Part
As rolled	2.7	20.8
As rolled	7.8	22.9

Sulphur prints were made from both the sound and unsound bars, but there was no distinction between them. These results appear to negative the conclusion that the annular rings in Dr. Stead's experiment broke pre-

maturely on account of sulphide segregation; they would have broken prematurely in any case, because blowholes or any other cavities in an ingot do not weld completely however much work may be put on to the material during forging or rolling.

The difference in behavior between the sound and unsound rings when drifted is a striking example of the disadvantages of blown or piped steel when stressed transversely. Gun tubes or gas cylinders are stressed more or less like the drifted rings, though they are, of course, not made into hollow tubes in the same way, and a welded blowhole would lie more favorably in either case than in the drifted rings. Gun tubes are not made from blown ingots, but blown ingots of mild steel are consciously made into billets and blooms for tubes and gas cylinders. Whatever commercial warrant there may be for this practice it would appear to be undesirable metallurgically, because blowholes induce spot segregates, and are undesirable mechanically because the incompletely welded cavity decreases the resistance of the steel to the working stresses.

There seems to be little doubt that sulphur has been blamed for defects due to other causes, and one wonders whether this segregate in particular is quite as harmful as it is made out to be. Take, for example, a piece of boiler plate which has laminated on flanging. The metallurgist applies a piece of acidified photo paper and produces a sulphur print (Fig. 11), showing that the defective part corresponds with a segregated area. According to text-books segregates are known to be harmful and high sulphur is reputed to cause red-shortness. The inference is easy, but the obvious explanation is, in many cases, not the correct one. It is true that sulphur compounds segregate in steel; so do phosphorus, and so also do carbon compounds. Of these the one most easily detectable, and almost invariably detected, by the method of sulphur printing is sulphur, but it is not therefore necessarily more culpable than either phosphorus or carbon. Further, in perfectly sound ingots (except perhaps very large ones) mass segregation is not possible. In such ingots as are practically sound, that is to say, in the best type of commercial ingots, mass segregates are found only where there are, or have been, cavities. The pipe in a piped ingot is lined with segregates and most cavities in a blown ingot are distinguishable in the forged blooms by the segregates which filled, or partly filled, the cavities in the ingot stage.

On reconsideration of the laminated boiler plate we must regard the dark center part on the sulphur print first as evidence that the ingot material, from which the defective article was made, was piped. The sulphide segregates enable us to draw this important conclusion, but whether the segregates as a whole have aggravated the defect depends on whether or not they have interfered with the chances of the pipe welding up. The facts are broadly the same with respect to spot segregates. In either case, that is to say, in relation to both axial segregates and spot segregates, the real cause of the trouble is either a pipe or a blowhole, and it is not reasonable to blame the segregate, which in the nature of things gets squeezed into these cavities, unless they interfere with welding. A series of five crucible steel ingots containing from 0.06 per cent to 0.17 per cent sulphur, but otherwise similar in composition, have been prepared in order to observe the influence of sulphur on the welding figure. The results are not ready for publication, but so far as they go they show that either in the rolled condition or in the hardened and tempered condition the high sulphur steels weld as well as, and on the whole rather better than the low sulphur steels. This, the general experience, leads the author to assume that of the segregates themselves, the least harmful, taking all things into account, is probably sulphur. By pure chance it happens to be easily detectable, and it is thus forced to turn king's evidence against its associates.

It would be something of a paradox if, presuming it could be entirely eliminated, sulphur were after all retained in, or purposely added to, steel in order to serve as witness. A sulphur print of an ingot section is one of the best records of the conditions under which an ingot is formed. It may indicate the casting temperature; whether top or bottom casting methods have been used; whether blowholes later filled up have existed earlier in the partly fluid ingot; and many other useful things to be the observant investigator. It is by far the most valuable means available for determining whether a forging became axially unsound on hammering or was made from piped material. It indicates at once the presence of spot segregates in forged or machined articles, and either facilitates the machining or suggests that the material should be scrapped, and finally it is an indispensable touchstone for the careful steel-maker who is making steel for such special purposes as rifle barrels. In nearly every case a sulphur print is less valuable as an indication of sulphur than of more harmful things of which its presence gives warning. A sulphur print is worth many sulphur determinations, and we could hardly spare the tell-tale sulphur altogether—we might then have to add sulphur to our steel as we know nothing so voluble, so reliable, and so harmless which would serve the purpose equally well.

Very few experienced observers will doubt the possibility of pulling apart welded faces in objects made from blown ingots. The author believes that it can be done, always providing the steel in which they occur is tough or can be made tough by heat treatment. Iron and steel lumps from which bars for machine knives are made are produced by casting very mild steel against the face of very hot tool steel. In the forged or rolled bar the inter-penetration is considerable (see Fig. 12). The two materials, however, can be pulled apart along the original faces. Three-ply steel, such as is used for making plough mould boards, is probably produced in a similar way, but the rolled sheet can be split and the iron core separated if the sheet be first toughened by hardening and tempering.

If a billet be drawn down at one end into a bar the unforged part of the billet may be worked up and down in fluid steel as it rises in an ingot mould, so as to heat the billet and drive off adherent gases. Such a billet may then be left in the upper part of the ingot in a highly preheated and clean condition. After forging that part of the ingot containing the billet may be rolled into flat bars and such bars may be tested. After this procedure a core corresponding with the original billet section may always be separated.

From these considerations one would expect that impact test-pieces cut from the center and from the edge of a flat bar rolled from an ordinary ingot would frequently give different results. Such differences actually exist, and moreover the difference is most marked when the bar is taken from that part of the ingot which is unavoidably least sound. An ingot cast wide end up and provided with an efficient feeder head is—omitting the feeder head—most nearly perfect in its bottom third and top fourth; it is less perfect in the center of its middle half. These facts are clearly discernible in a sulphur print made from a sound ingot which has been sectioned longitudinally through its center. Quite in accordance with the sulphur print indications one finds that variations in impact test-pieces cut from the edge and center of flat bars are greater or less, according as the bars have been made from the middle or one or other extremity of the ingot.—"Engineering," May 6, 1921.

THE PREVENTION OF SEASON CRACKING IN BRASS BY THE REMOVAL OF INTERNAL STRESS.*

By H. Moore, O. B. E., B. Sc., F. I. C., and S. Beckinsale, B. Sc., A. I. C.
Communication from the Research Department, Woolwich.

In the work now to be described specimens maintained in a state of stress by external constraint were annealed at temperatures from 200 degrees to 325 degrees C., and the tensile stress, the amount of which was known before treatment, was determined after the treatment had been applied.

The work was confined to 70:30 brass substantially free from impurities. Strip cold-rolled from the fully annealed condition to Brinell hardnesses of 120, 160 and 200, respectively, without further annealing, were used. The strip was well prepared from one ingot made from electrolytic copper and zinc, both of high purity, and contained 71.2 per cent of copper. The Brinell hardness of each strip used was determined with a ball of 1 mm. diameter and a load of 10 kg., by means of the machine designed in the Research Department, Woolwich, for the determination of the hardness of thin material, and described elsewhere by one of the authors.†

When a flat strip is bent into the form of an arc of a circle, of radius relative to the thickness of the strip such that the elastic limit is not exceeded, the external surface is maintained in a state of uniform tension, the intensity of which may be calculated from the well-known bending formula.

$$p = \frac{E\,t}{2\,r}$$

where p = tensile strength at the outer surface.
E = Young's modulus.
t = thickness of strip.
r = radius of curvature.

E was determined as 7,300 tons per square inch for 70:30 brass cold-rolled to 160 Brinell hardness. That the stress does not exceed the elastic limit may be established by return of the strip to its original straight form on release.

The condition of such a strip, maintained in the bent form by external constraint, is similar to that of an internally stressed brass object the external surface of which is in tension, and it may be assumed that the stresses will be reduced by low-temperature annealing in the same manner as in the case of internally-stressed objects. It would appear that the reduction of stress by heating is due to slight yielding of the material taking place at the higher temperature.

When a strip bent in this manner is heated for a suitable time above a certain limiting temperature, allowed to cool, and then released, it will remain in the bent form or return more or less completely to its original straight form according as the initial stress has been entirely removed or merely reduced to a smaller or greater extent. If the form of the strip on release is an arc of a circle having a radius r_1 greater than the radius r

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* Abstract of a paper read at a joint meeting of the Faraday Society with other Institutions, April 6, 1921.

† H. Moore, "A Small-Ball Hardness Testing Machine," Proceedings of the Institution of Mechanical Engineers, 1921 (January), page 51.

of the arc to which the strip was originally bent (see Fig. 1), the tension p_1 after treatment and before release may be calculated by application of the simple bending formula

$$p_1 = \frac{E\,t}{2}\left(\frac{r_1 - r}{r_1\,r}\right)$$

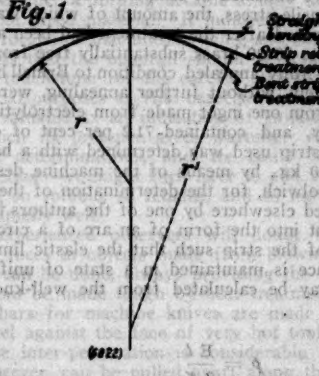


Fig. 1.

p_1 is the remaining stress in the surface after treatment, before release.

The procedure adopted was to maintain in the bent form a length of strip, elastically bent to the form of an arc of a circle, apply a given heat treatment, release after cooling and measure the radius of the arc which the strip assumed. The initial stress and remaining stress (before release) were determined in the manner indicated above. In all cases the form assumed by the strip on release was that of an arc of a circle.

In order to obtain different initial tensions with a constant radius of arc three different thicknesses of strip (0.02 inch, 0.015 inch and 0.01 inch of each of the three different hardnesses (Brinell numbers 120, 160 and 200) were employed. It was found by trial that a circle of 7 inches diameter was the smallest to which the softest brass used (120 Brinell number) could be bent when of the maximum thickness used (0.02 inch), without approaching too closely the elastic limit, beyond which permanent deformation would result.

The lengths of strip were attached at each end to steel formers having a diameter of 7 inches, by means of small blocks having an internal radius of 3.5 inches. The strips were 6 inches long and 1 inch wide. In order to confirm that in no case was permanent deformation produced by the initial stress a number of strips of each hardness and each thickness were clamped on the formers and maintained at atmospheric temperatures for a length of time equal to that of the longest treatments. On release all returned to the straight form.

The heat treatments were carried out in a bath of molten carnauba wax (described in the appendix) the temperature of which was maintained by a thermostat within $\pm$ 0.5 degrees C. of the temperature aimed at. The protracted annealings (exceeding 24 hours) at 200 degrees and at 225 degrees C. were carried out in an air oven uniformly heated by means of a

jacket of molten carnauba wax. In this oven the temperature variations were somewhat greater than in the bath, but in no case did the temperature rise more than 4 degrees C. above that stated.

The results were given in tabular form and typical results shown graphically in the paper. The effect of similar treatments on the Brinell hardness of identical pure 70:30 brass strip cold-rolled to Brinell hardnesses of approximately 90, 120, 165 and 200 was also indicated. In these tests, also made in the machine referred to above, a ball of 2 mm. diameter and a load of 40 kg. were used.

Mechanism of the Removal of Stress by Low-Temperature Annealing.—It has been pointed out above that the removal of stress in a specimen or object which remains unaltered in form can be brought about only by the occurrence of plastic flow. It is perhaps more correct to state that when stress is removed in the manner under consideration the elastic deformation becomes permanent deformation, the extent of the removal of stress depending on the extent to which the elastic deformation becomes permanent. It follows that stress can be reduced by heat treatment only by a lowering of the elastic limit below the initial stress. The lowering of the elastic limit may be, and in suitable treatment is, only temporary, the elastic limit rising again on cooling after heat-treatment; it will be seen indeed that the ultimate effect of low-temperature annealing is in many cases to raise the elastic limit.

Accordingly it may be stated that the removal of internal stress in 70:30 brass by low-temperature annealing depends on the changes in the elastic properties which occur at temperatures in the range of about 200 degrees to 300 degrees C. The authors have carried out a number of experiments for the investigation of these changes and although the work is incomplete at present they consider the results so far obtained to be of sufficient interest to justify their inclusion in this paper. The experiments have been carried out on specimens cut from cold-rolled 70:30 brass plate of about 170 Brinell hardness and have included:—

(1) Determinations of load-extension diagrams in tensile tests made at 275 degrees and 300 degrees C., in comparison with similar determinations made at 17 degrees C.

(2) Determinations of the amount of permanent set under given loads maintained for given lengths of time at a temperature of 275 degrees C.

(3) Determinations of load-extension diagrams in tensile tests at 17 degrees C. after treatments at 250 degrees, 275 degrees, 300 degrees and 325 degrees C. to determine how far the changes observed in (1) were temporary or permanent.

The Ewing extensometer reading extensions to 0.000008 inch on a gauge length of 2 inches was used for tests carried out at ordinary temperatures, while a specially-constructed extensometer described in the appendix was used for the high-temperature tests. The electric resistance furnace used in (1) and (2) is also described in the appendix. All tensile tests at raised temperatures and treatments under load were carried out in an Amsler 5-ton vertical testing machine.

The actual copper content of the brass used was 72.6 per cent. The thickness of the plate was 0.75 inch; the test pieces were cut longitudinally, were circular in section and were 0.504 inch in diameter throughout the acting length.

1. In the determination of load extension diagrams at high temperatures the test-piece, enclosed in the electrically-heated tube furnace, was placed in the vertical testing machine, the extensometer was set and the temperature, measured by means of a thermocouple fixed in contact with the test-piece, was raised to the required degree. When steady conditions were indicated by steady readings on the extensometer dials, load was applied

in increments of 1 ton per square inch and extensometer readings were taken at each increment. Each test occupied about 15 minutes from commencement of loading. After a sufficient number of readings indicating a considerable departure from proportionality and thus a large permanent set, the extensometer dials were removed and the test-piece was broken at the same temperature.

Duplicate tests carried out at each temperature gave results agreeing very closely with the figures in the table. It was found that corresponding departures from proportionality of strain to stress occurred at much lower loads at 275 degrees C. than at 17 degrees C. and at lower loads still at 300 degrees C. The amount of permanent set, as judged from these results, does not, however, appear to be sufficient to account for the extent of reduction of stress which occurs at the temperatures tried, in the annealing of strips initially stressed by bending.

2. In the determinations of permanent set under loads maintained for given lengths of time at 275 degrees C., a gauge length of 4 cm. was marked off by fine crosses on test pieces of the same dimensions as used in (1) (diameter 0.564 inch over a length of 2.25 inches). This was measured with a measuring microscope reading to 0.00004 in. The test-piece, enclosed in the furnace, was gripped in the testing machine, and was heated to 275 degrees C. Uniform heating and a steady temperature were attained in 30 minutes from commencement of heating. The given load was then applied and maintained for the stated time while the temperature was held at 275 degrees C. Each test was carried out on a fresh test-piece. Useful indications of the rate and amount of extension occurring during the test were given by the load-indicating dial of the testing machine and by the adjustments of the straining screw required to maintain a constant load. The treatment of the test-piece to which no load was applied was carried out in the testing machine in the same way, but with the lower grip disconnected.

It is evident that in the conditions of these experiments extension under a constant load continues for some time at temperatures similar to those at which internal stress is reduced, and that in a given time of maintenance of load permanent extension begins at very small loads and increases with increase of load. Indeed a slight increase of length occurs on treatment for 30 minutes at 275 degrees C. with no load on. This has been confirmed by repeat experiments. It would appear that the changes occurring in hard-rolled 70:30 brass when submitted to this treatment, changes which include an increase in hardness and in range of elasticity, are accompanied by a slight alteration in dimensions. This is a matter which demands further investigation.

The gradual flow which takes place under load at the temperatures under consideration explains the gradual reduction of stress which is observed when internally or similarly stressed specimens are heated at these temperatures.

3. It is clear that at temperatures in the range 200 degrees to 300 degrees C. the elastic limit of brass cold-rolled to different degrees of hardness is lowered. To determine to what extent elasticity is recovered on cooling, pieces were maintained for 1 hour at 250 degrees, 275 degrees, 300 degrees and 325 degrees C., allowed to cool and machined to the form of tensile test-pieces identical with those described in the two preceding sections; load-extension diagrams were then taken, the Ewing extensometer being used. The results showed that after treatment for times not exceeding 1 hour at temperatures below 300 degrees C. the range of elasticity in tension is certainly not lower than before treatment. Treatment at 250 degrees and at 275 degrees C. has considerably raised the limit of proportionality.

(The Brinell hardness tests were made with a load of 1,000 kg. and a ball of 10 mm. diameter. A test-piece treated at 250 degrees C. was initially somewhat softer than the others; for this reason it is considered that the apparent slight fall in maximum load after treatment at 250 degrees C. should be disregarded.)

In connection with the work described above in Sections 1, 2 and 3 of this portion of the paper, the authors are aware that questions arise as to the real nature of the elastic properties of cold-worked material, the definition of "limit of proportionality," "yield-point," and other terms used, hysteresis in the repeated loading of such material, the mechanism of the process of "recovery" of elasticity brought about by low-temperature annealing, and other fundamental problems on which present knowledge is incomplete. The authors have avoided the consideration of these questions in the present paper as being unnecessary in their presentment of experimental facts and of a working explanation of the reduction of internal stress by slow-temperature annealing.

Summary and Conclusions.—The reduction of stress by low-temperature annealing has been studied in 70:30 brass cold-rolled to 120, 160 and 200 Brinell hardness, in elastically bent test-strips of each hardness, initially stressed to 11 tons, 17 tons and 22 tons per square inch maximum tension at the surface. The temperatures used were 200 degrees, 225 degrees, 250 degrees, 275 degrees, 300 degrees, and 325 degrees C. Thus the effects of four variables, temperature, time, hardness and initial stress were separately determined. The results may be summarized as follows:—

(1) The rate of reduction of stress is fairly rapid at 200 degrees C. at first, but becomes very slow when the stress has been reduced to one-half to one-third its initial value and important stresses remain even after treatment for 24 hours or longer.

(2) As the temperature is raised the rate of reduction of stress increases but shows the same characteristic of slowing down as the stress falls. At 300 degrees C. a very much shorter time is required to reduce the stress to a given figure than at 200 degrees C. and the remaining stress is much lower after a given time at the higher temperature in brass of the same hardness.

(3) The higher the initial stress the higher is the remaining stress after a given treatment, in brass of the same hardness, although the amount of stress removed is greater the higher the initial stress.

(4) The higher the hardness of the brass the lower is the remaining stress after a given treatment and for a given initial stress. In other words, the harder the brass the more rapidly is a given initial stress reduced at a given temperature.

(5) A large reduction in the amount of stress is brought about by annealing conditions (temperature and time) which raise the hardness of cold-worked 70:30 brass, but treatments which result in some reduction of hardness are necessary to bring about complete removal of stress.

(6) At temperatures in the range 200 degrees to 300 degrees C. slight plastic flow occurs in cold-worked brass at low stresses. The amount of flow is greater as the temperature is higher, the stress greater and the time longer for which the stress is maintained.

(7) The reduction of internal stress by low-temperature annealing is dependent on the plastic flow mentioned in (6).

(8) This plastic flow involves a lowering of the elastic limit at the temperatures at which it occurs. This reduction of elastic limit is, however, not permanent provided that the treatment is applied for a time not exceeding 1 hour at 275 degrees C. or certain limiting times which are much

longer at lower temperatures. The limit of proportionality and elastic limit are raised considerably by treatments which do not exceed these limits of temperature and time.

The results obtained and summarized above confirm and extend the earlier work of the authors given in the Proceedings of the Institute of Metals. By a suitable low-temperature annealing internal stress in 70:30 brass may be reduced to such an extent as to eliminate the risk of season-cracking, without injurious effects on the mechanical properties of the brass. By suitable choice and reasonably accurate control of the temperature and time of the treatment which reduces the stress to a safe limit the mechanical properties may indeed be improved.

Preliminary experiments carried out by the authors on 60:40 and other brasses suggest that this general conclusion can probably be extended to all industrial brasses. It would therefore appear that season-cracking of brass resulting from internal stress is a type of failure which should be entirely eliminated by the application of a suitable low-temperature annealing to the manufactured article after the final cold-working operations.

APPENDIX.

1. WAX BATH USED FOR LOW-TEMPERATURE ANNEALING.

This apparatus, which is suitable for treatments at temperatures in the range 200 degrees to 325 degrees C., consists of an electrically-heated bath containing carnauba wax, a thermostatic control and a temperature recorder, together with controlling rheostats, ammeter, &c.

Bath.—This is a lagged sheet steel cylindrical vessel. The heating element consists of two square uralite frames on which nichrome wire of No. 18 gauge is wound. These frames, connected in series, are fixed horizontally near the bottom of the tank, and are separated from the tank and from each other by suitable insulation. Three inches above the frames a perforated sheet steel false bottom is fixed. Current is supplied from a 250-volt circuit.

Thermostatic Control.—This is in principle a mercury thermometer, having a large steel bulb and a fine steel tube lined with a closely fitting silica capillary tube. A nickel wire passing into the silica tube is fixed with the lower end at any desired height in the tube. Contact of the mercury with the wire completes an electrical circuit, which by means of a solenoid actuates a mercury switch and breaks the heating circuit. When the temperature falls the solenoid circuit is broken by the mercury leaving the wire, and the heating circuit is restored.

Temperature Measurement, &c.—Standard mercury thermometers were used and autographic records were taken by means of a Cambridge thermograph. Throughout all treatments the molten wax was continuously stirred by means of a motor-driven rotating paddle.

The formers to which the strips were attached for the treatments described in the paper were suspended in the molten wax.

2. EXTENSOMETER FOR USE AT RAISED TEMPERATURES.

This instrument, constructed in the Research Department, Woolwich, is similar to that described by H. J. French (United States Bureau of Standards). The extensometer consists of two light steel frames each rigidly fastened to a steel yoke by two rods, adjustable in length for attachment to test-pieces of different length. The yokes are clamped to the specimen by means of four hardened high-speed steel screws.

The flanges on the upper frame carry two Ames dials reading to 0.0001 inch, the pistons of which make contact with two adjustable polished steel platforms carried on the lower frame.

With an extensometer of this type uniform symmetrical stretching of the test-piece causes an equal movement of each platform from its corresponding dial. If the stretching is accompanied by any twisting or bending of the test-piece the dial readings will not be equal, and the reading of one dial may even be negative. Accordingly the extension is taken as half the algebraic sum of the readings of the two dials. It may be noted, however, that in the tests which have so far been made with this instrument the differences between the readings of the two dials have been relatively quite small. As may be noted from the readings given in the paper, satisfactory agreement between this extensometer has been obtained in tests made on identical test-pieces at atmospheric temperature.

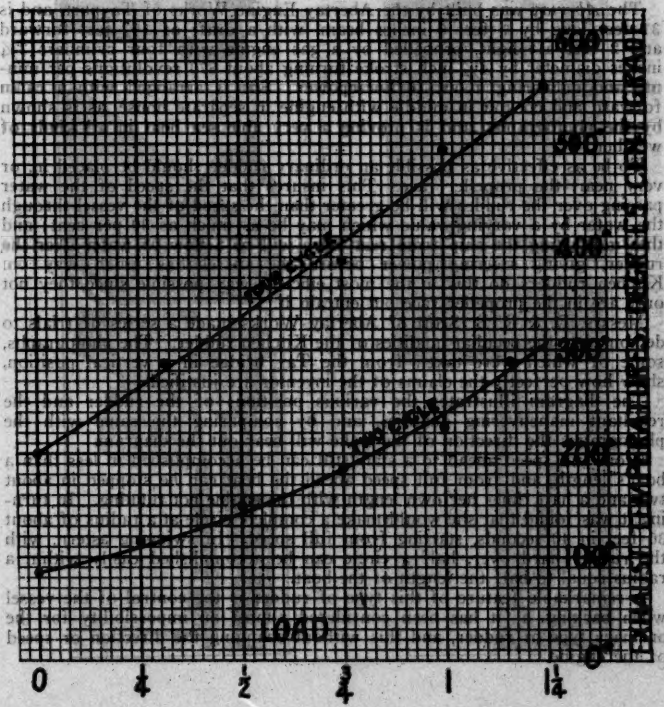
3. FURNACE.

The test-piece and extensometer yokes occupied the central part of the furnace, the ends of which were loosely plugged with asbestos wool. The arms of the extensometer remained at atmospheric temperature.—"Engineering," May 27, 1921.

MARINE ENGINES.

EXHAUST TEMPERATURES IN FOUR AND TWO CYCLE ENGINES.

It is still little admitted that the two cycle engine as regards the height of the exhaust temperatures is much more favorable than the four cycle



engine. In the accompanying diagrams the exhaust temperature of the large four cycle and two cycle engines are represented as functions of the load.

The results were obtained from trials on engines of the Sulzer Brothers in Winterthur who, as is known, build both four and two cycle engines and therefore have learned to correctly estimate the advantages and disadvantages of both systems. The temperature of the exhaust gas in the two cycle engine is, according to the load in round numbers 120 degrees to 250 degrees C. lower than that of the four cycle engine. It remains, with a temperature of approximately 300 degrees at overload, within practically easily controllable limits. Exhaust pipes in large two cycle engine plants therefore never attain the temperature of hot steam inlet pipes in steam engines.—From "Schiffbau," No. 2, page 68.

SOME OBSERVATIONS ON THE KITCHEN RUDDER.

By R. Z. Dickie.

Considerable interest is being manifested in the launch *Abertay*, which is being operated on the Firth of Tay, Scotland, as a ferry between Tayport and Broughty-Ferry, as she is equipped with a Kitchen rudder to demonstrate its maneuvering qualities with an engine which operates in the ahead motion only.

The *Abertay* was built by the Abertay Engine Works of Tayport, and is 31 feet long by 7 feet 6 inches beam, with a draft of 1½ feet forward and 2 feet aft light, propelled by a gas engine with four cylinders 3¼ inches diameter by 4½ inch stroke turning about 870 revolutions per minute and delivering about 16 horsepower. She is arranged with a cabin forward and cockpit in middle with engine in separate house, as is shown by the photographs, and is proving a very able sea boat in all kinds of weather.

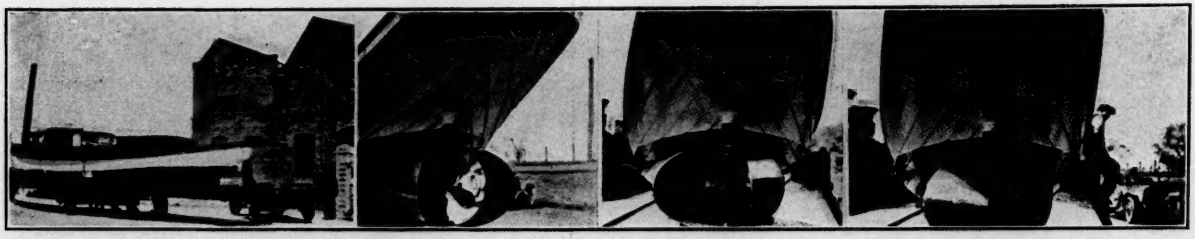
To be as effective as possible an ordinary rudder should be placed in, or very near, the propeller race. This insures that the speed of the water passing over the rudder will be greater than the speed of the vessel through the water by a varying value which may be as much as 30 per cent; and that in starting the ship from rest there will be a flow of water over the rudder, giving steering power before the vessel has much way on. Kitchen rudders do this in the most efficient way possible since they not only are in the propeller race but entrain it.

Messrs. D. & R. B. Scott of Abertay Works made a series of trials to demonstrate the peculiar qualities of the Kitchen rudder. The photographs, some of which were taken from the Tay bridge in a vertical position, show how perfectly the claims of the inventor are justified.

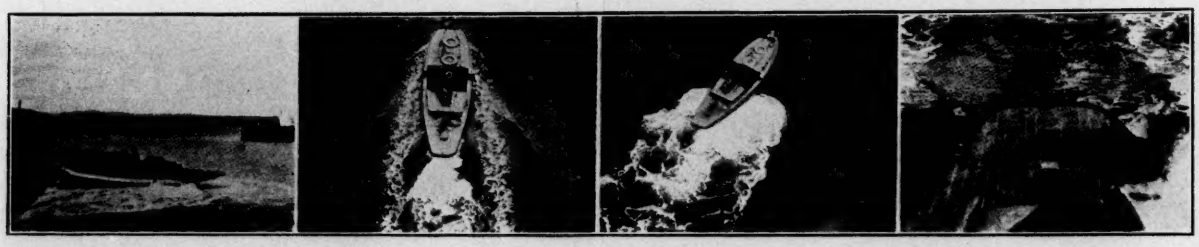
The diagram illustrates the various positions of the rudder and the resultant maneuvering qualities, and by comparing the same with the photographs the direction of the wake will bear out the diagram.

From full speed ahead to a standstill can be accomplished in less than a boat's length, and from full speed ahead the boat can be stopped in about two and a half times her own length with the engine not running. In turning it was found that she would make a complete circle at a radius of about 30 feet in 40 seconds, starting from full speed. In swinging astern, with the rudder hard over, half a circle can be accomplished easily within a radius equal to half the length of the boat.

A remarkable feature of this type of rudder is the control of the vessel when backing, as it has been considered almost an impossibility for the ordinary type of rudder, and this without changing the direction or speed of the engine.



LEFT TO RIGHT.—LAUNCH "ABERTAY" LEAVING SHOP FOR HER TRIALS. SHOWING SIDE VIEW OF KITCHEN RUDDER. VIEW OF STERN SHOWING RUDDER IN STRAIGHT AHEAD POSITION. STERN SHOWING RUDDER HARD APORT. STERN SHOWING RUDDER CLOSED IN HARD ASTERN POSITION.



LEFT TO RIGHT.—"ABERTAY" MANEUVERING AT DOCK SWINGING TO PORT. VIEW FROM TAY BRIDGE LOOKING DOWN ON LAUNCH AS RUDDERS ARE CLOSED TO STOP. VIEW LOOKING DOWN SHOWING LAUNCH ALMOST STOPPED. NOTE ACTION OF RUDDER ON WATER. VIEW OF WAKE FULL SPEED AHEAD. ABOVE.—SIDE VIEW OF LAUNCH BROUGHT TO STOP POSITION WITH ENGINES FULL SPEED AHEAD.

A valuable asset in favor of the rudder which cannot be overlooked in the case of a high speed engine requiring a propeller with a low pitch is that the efficiency is not lost in backing as the engine always runs in one direction while with an ordinary reverse gear the low pitched propeller would be very inefficient for backing if properly designed for going ahead.

Experiments with Kitchen rudders have shown a gain in propeller efficiency when the rudder is partly closed. With the same power applied there has been a uniform gain in speed of approximately five to ten per cent over that obtained with the ordinary rudder, but the exact amount gained with the rudder closed is a matter of experiment, for no two boats seem to give the best results with the rudder closed the same amount.

In conclusion it might be added that when a boat is equipped with a Kitchen rudder and in the hands of a skilful operator the control is almost absolute.—"Pacific Marine Review," November, 1921.

REFLECTIONS ON NAVAL DISARMAMENT.

The suspension of work on the four British ships is the most important development which has occurred since the American plan was announced to the world. It was, we believe wholly unexpected. Few people imagined that a pious avowal would so rapidly be translated into a fact, and there is still much perplexity as to the reason which inspired the hasty action of the government. It is considered unlikely that the proposals will be accepted without modification. Amidst all the enthusiasm which they have evoked it is well to remember that they were drawn up by American statesmen and American naval officers, who were doubtless guided as much by considerations of national interest as by motives of pure altruism. That being so, it would be absurd to pretend that we in this country have no right to criticize or question a single item in the program. It is not only our right, but our manifest duty to scrutinize with particular care a plan which aims at determining British naval policy over a long period of years, for the Navy is still the Empire's first line of defense, and sea power—in whatever sense the phrase may be interpreted—remains, as it has ever been, the cardinal condition of the Empire's existence. If therefore, it were found on closer investigation that any part of the plan was incompatible with the maintenance of an adequate measure of sea power, it would be the duty of those representing the British Empire to suggest the necessary modifications. In these circumstances we cannot but feel that it would have been wiser to continue work upon the ships if for no other reason than that the orders for material would have reduced unemployment.

There is little reason to suppose that drastic amendment will be necessary. The limits of capital ship strength appear to be fair and reasonable to all three parties, with the possible exception of Japan, and it is mainly in regard to the proposed allotment of cruiser and submarine tonnage that grounds for criticism exist. Assuming that cruisers are necessary for the protection of sea-borne commerce—and nothing in the experience of the late war justifies the contrary view—then it is obvious that the British Empire, with its unprecedented volume of ocean trade and its dependence on the safety of the marine highways, could legitimately lay claim to a larger proportion of such vessels than is needed either by America or Japan. It remains to be seen whether the American naval authorities are willing to concede this claim, which in our judgment must be firmly upheld. In the opening stages of the late war we were seriously hampered by the dearth of fast cruisers, and had the enemy displayed more enterprise and boldness he might have made us pay heavily for our neglect to provide a sufficient number of high-speed cruising ships. Since we have no guarantee that in

a future war our commerce would be immune from the attentions of surface raiders, like the *Emden* and the *Karlsruhe*, the risk must be minimized by keeping the cruiser establishment at the standard of strength recommended by our naval advisers. It seems to us neither prudent nor feasible for this country to surrender the right to take such precautions as may be deemed requisite for the protection of its floating commerce, considering the vital part which that factor plays in the national existence. Exception has also been taken to the submarine proposals, but for very different reasons. In this case the British view is not that America offers too little, but that she offers too much. It is a fact that the suggested figure of 90,000 tons as the limit of displacement for the British submarine flotilla is greatly in excess of our present establishment, so that in this type of vessel it would be a case not of "scrapping," but of new construction, it we wished to attain the prescribed standard of strength. We infer from their very liberal allowance of tonnage that American naval experts set a high value upon the submarine, and in this they are doubtless right. It is a formidable weapon both for defense and offense, but as the nation which suffered most from its employment in the latter capacity, Great Britain regards it, not without reason, primarily as an instrument of offense, and one to which she is peculiarly vulnerable. No secret is made of the fact that our Government would welcome the total elimination of the submarine. If that were agreed to it would be considered, in this country at least, by far the greatest achievement of the Washington Conference. We fear, however, that the submarine will not be got rid of so easily. When tentative proposals for its abolition were put forward at the Peace Conference they met with such uncompromising opposition from several quarters that they were not pressed. It was pointed out that the British objection to the submarine was not wholly disinterested, that it was essentially the weapon of the weaker maritime powers, and that its disappearance would simply confirm the supremacy of the dreadnought, which only the richest states could afford to build. But although these arguments would not be difficult to refute, the real point is that since American naval opinion is partial to the submarine, it would be useless to discuss the question of abolition. Nor, as we observed last week, would much practical value result were a limit to be set to the dimensions of future submarines. It would be possible, without exceeding a very modest displacement, to endow a submarine boat with a cruising radius that would enable it to act as a commerce destroyer on the high seas. The radius of a submarine is largely determined by the physical and moral qualities of its crew. There were many instances in the late war of boats considerably smaller than 1,000 tons undertaking ocean voyages up to 7,000 sea miles, a performance that would have been thought incredible before the war.

If we fail to secure at Washington a substantial reduction of the amount of submarine tonnage originally proposed in the American scheme, and if, at the same time, the Conference leaves its participants a free hand in respect of submarine dimensions, the next few years will assuredly witness the intensive development of this type of craft. The United States has just begun work on two fleet submarines which are said to have a surface displacement of approximately 2,000 tons and a sea endurance of 10,000 miles. Japan is credibly reported to be building even larger boats, which are to be of exceptionally high speed, judging from the power of their Diesel engines, which have been ordered from a well-known firm in Europe. And on November 1st the keel was laid at Chatham Dockyard of what is described as the largest submarine ever designed for the British Navy. To these facts must now be added the authoritative report that two of the American battle-cruisers, whose construction as such will be arrested under

the limitation scheme, will be completed as aircraft carriers, each with a speed of 33 knots and accommodation for 100 bombing, torpedo and fighting planes. It would appear, therefore, that if American naval opnion is somewhat doubtful as to the possibility of maintaining perpetual peace on the seas, it is fairly confident that in any future naval campaign submarines and aircraft will play the leading roles.—"The Engineer," November, 25, 1921.

MacREADY'S ALTITUDE FLIGHT.

In a special dispatch to the New York World, Lieut. John A. MacReady tells the remarkable story of his world's altitude record flight as follows:

"It wasn't a record I was after when I started out on the flight which ultimately resulted in upsetting Major R. W. Schroeder's altitude record, but to test a new propeller and additional equipment on the Moss supercharger, which has been experimented with extensively at Cook Field.

"One of the functions of the engineering division is the development of aeroplanes and equipment for military use at high altitude. Continual effort is being made to develop planes which will be better than the planes of other nations for combat or observation work at extremely long distances over the ground.

"My record flight yesterday was a test flight and the record was incidental to the very valuable engine, propeller and supercharger data which were obtained.

DAY WAS IDEAL FOR TRIP.

"The day was ideal—warm and clear—just the sort of day on which the best results are obtained at altitude. The warmer the day is the better instrument results are obtainable, because the pressure and density corrections do not bring the final height so low after all instruments are checked up and calibrated.

"After a good night's sleep to insure the best physical condition for the flight I arrived at the field and began at once to make immediate preparations toward insuring reliability of all factors which are necessary to obtain safety and comfort when undergoing the hardships which one encounters in this work. The mechanics had stripped the plane of all excess weight by cutting out unnecessary equipment and by reducing the ordinary gas capacity.

UNNECESSARY GEAR STRIPPED.

"The supply of oxygen was also reduced for the reason that in all previous flights I had flown with an observer, thus making it necessary to have an almost double supply of oxygen. Roy Langham had been my observer on all previous flights of altitude, but in this particular case it was not necessary for him to take readings, as the data to be obtained was such that I could get it myself with very few readings and observations.

"The plane climbed remarkably well considering the original slow speed of the propeller. This propeller was a very large one, having a ground speed of only 1,100 revolutions a minute, whereas most propellers have a speed around 1,600. The density of the atmosphere decreases with altitude, and a propeller turning slowly at the ground would have great speed as the plane rose more and more into the lighter air. On one previous flight the propeller speeded up from 1,450 revolutions per minute at the ground to 2,400 revolutions per minute at 34,000 feet, which was the ceiling of this particular flight; and on another occasion, when attempting an altitude

flight, the propeller flew off the plane in the air, knocking off the air speed indicator, which was attached to the wing strut of the plane, and hitting the strut, because of this excess speed.

HAD BEEN TRYING FOR YEARS.

"For more than a year I have been attempting to reach an altitude higher than has previously been made, but on all occasions something has broken while taking the plane up on the flight, and it has been necessary to descend in trouble. Oil pipes and bearings have broken in the engine, parts of the supercharger have flown off in the air, and the propeller has been lost in flight. The majority of these flights have been test flights around 25,000 feet for the purpose of obtaining data.

"Everything was functioning properly during the early part of this flight on which the record was broken. As altitude was gained I became more confident that success would be attained because in all flights I have made with the supercharger none had shown such excellent functioning during the early part of the trip. All the mechanisms were working in excellent order; I began using oxygen at approximately 20,000 feet, using it very sparingly at first in order that there would be sufficient in case of emergency.

"I was well protected against the cold. Over my military uniform was a heavy suit of woolen underwear and over this was a thick, heavily padded leather covered suit made of down and feathers. On my feet were fleece lined leather moccasins. My hands were covered with fur-lined gloves. My leather head mask was lined with fur and an oxygen mask was attached to it.

"The goggles were separate and were placed on the outside of the head mask. The inside of the goggles were covered with a film of secret gelatine compound which was painted over the glass. This gelatine compound is for the purpose of keeping the ice from forming on the inside of the goggles. It is supposed to function in this respect to a temperature of approximately 60 degrees F. below freezing.

"In addition to the standard oxygen equipment of five flasks containing together a pressure of 1,300 pounds, I had an additional emergency flask containing a pressure of 1,500 pounds, which would lead directly from the flask through a tube into my mouth. At 10,000 feet I passed through a stratum of clouds and at 20,000 feet passed through another stratum which somewhat obscured the ground and hindered me in locating my position with respect to the geography of the ground. I did not feel any ill effects whatever until well above 30,000 feet, as I was receiving plenty of oxygen and was warm enough, but as the altitude above 30,000 feet was reached a slight slowing up of one's senses and faculties was noticeable and this slowing up increased as altitude above this height was gained. Any exertion calls for oxygen. The faculties do not function normally.

"If I would stoop over to make some adjustment I would feel the need for oxygen. This need is manifest by the objects on the ground and the instruments in the cockpit becoming dim and shaky. Whenever I felt these effects I would turn on a little more oxygen, which was flowing well and freely. I was worried at no time until approximately 39,000 feet was reached.

"At this altitude ice from my breath within the mask must have clogged the oxygen pipe, for the reason that I felt the force becoming diminished and began to feel very bad effects from its lack. I tried to blow this out and did succeed in getting a taste of the ice, but could not act quickly enough to clear the entire tube, so swung over to the emergency flask and

tore a small plaster from the side of my mask, placing the tube through this aperture directly into my mouth within the mask and in an instant was feeling comparatively normal.

"I was supercharging to sea level condition at 40,800 feet (indicated) and expected that flying on the engine alone I would probably reach 7,000 or 8,000 feet more, but this was not the case. As soon as sea conditions were lost on the engines the power was quickly diminished and I could only get 100 feet more altitude, which was the absolute ceiling of the plane.

"While hanging suspended at an indicated altitude on the dial of 41,200 feet the plane swung and rolled and very little action on the controls could be obtained. The controls were almost useless, as there was not enough sustaining surface to move the plane in the direction that was desired.

"It was at this point practically without control, and I held it there for almost five minutes before I was absolutely certain it would go no further. When assured of this I pulled the throttle slightly back in order to glide down, but even with this small movement the bottom seemed to drop out of the plane and down it quickly went toward earth.

ICE NOW BLINDS GOGGLES.

"Before I could make the proper readjustments the engine and radiator had cooled so quickly that there was no warmth in the cockpit from the radiator pipes running through it to warm the pilot, and as a result it became much colder in the cockpit, resulting in ice forming on the inside of my goggles, making me almost blind for the instant, and I could not handle the plane correctly for a short time. I was feeling weak and groggy and was afraid of passing out completely.

"My mind was not active and I could not think fast and correctly. However, I knew that by reducing my altitude I would come back to normal and be able to make all adjustments if I could get down lower. I let the plane come down almost any way in order to get it down where I could think correctly.

"I neglected to say that just before or while I was at the supreme altitude that I remembered that Major Bane had asked me to look above me while at the great height and see if I could see the stars or any constellations and note the appearance of the sky and any peculiar features which might exist. This I did.

"I was unable to see any stars, but one fact was brought forcibly to my mind, and that was the fact that the atmosphere was extremely bright and light.

SKY LIGHT BLUE, NOT DARK.

"Instead of being a dark blue, as is the case at the ground, the sky was very light in color, with but a slight blue tinge. There is far more sunlight, and it is far brighter at this altitude than closer to the ground. Another point that I neglected was the fact that while climbing to the ceiling I was circling around the City of Dayton, gradually increasing the size of circle as altitude was gained in order to always be within gliding distance of the field or of Dayton. This circle as altitude was gained increased its radius until at the top of radius of about sixty or seventy miles. I attempted to keep this circle round, with Dayton as the center, but strong, heavy winds tended to drift me in various directions. I do not know what these directions were for the reason that my mind was not sufficiently acute to be able to know all these facts.

"The original part of the return to earth was made in a very confused state, but as altitude was lost, elements of consciousness became clearer,

and at about 20,000 feet I was beginning to feel normal again. Here I changed goggles and continued my glide toward earth, flying around at 20,000 feet for a period of approximately twenty minutes, in order that the change in condition from my altitude to the ground would not be too sudden.

INVENTOR CAME WITHIN 200 FEET.

"I landed and taxied to the line where the boys in the hangars and on the field were waiting, curious to know the results of the flight. A pool had been made on the hangar, the different mechanics betting on the altitude. Dr. Moss inventor of the supercharger, won the pool with a guess of 41,000 feet. The actual calibrated altitude of the flight, as indicated in the barograph charts, was 40,800 feet, which is the highest altitude any living being has made.

"I wish to again call attention to the fact that this record flight is but merely incidental in the chain of development of superchargers undergone by the engineering division of the air service and is the finished and perfected work of this development. Superchargers are in such a state now, due to the developing department under Major T. H. Bane, Capt. G. E. A. Hallett, Dr. S. A. Moss and Adolph Berger, chief of the power plant at the field, that they should be fairly consistent at extremely high altitudes in the future.

"Yet, in my opinion, altitude records will not be gauged by the superchargers themselves, but by the physical limitations of the pilots. To get much higher, greater provision of some method will have to be made or it will be impossible for a pilot to sustain the extreme hardship."—"Aerial Age Weekly," October 17, 1921.

STAINLESS STEEL AND ITS PROPERTIES.

In our issue of June 3rd we published an article in which we summarized the case for and against stainless steel, as disclosed in the replies to letters which we had addressed to a considerable number of engineering and other firms who had used or tried or were using the material. The position disclosed, it may be recalled, showed a wide diversity of opinions regarding the utility of this variety of steel. While some firms expressed themselves highly in favor of it, one or two were emphatically opposed to its use—at least for their particular class of work. An extensive range of minor uses of the material was recorded, but with the exception of table cutlery—there was no instance reported in which it was being used on a large scale, although it was not unlikely in some cases that it might come into extensive employment if preliminary tests then being carried out gave satisfactory results.

Our previous article may be said to have summarized the position from the user's standpoint, although by way of conclusion we gave the views expressed by the original discoverer of stainless steel, Mr. Harry Brearley, on certain points raised by some of our correspondents. We now propose to go further into the case from the makers' standpoint, for it seems undoubtedly to be the fact that some part at least of the difficulties and troubles encountered in the use of stainless steel is to be ascribed to the lack of a complete understanding of its nature on the part of certain users.

It is important at the outset to note that the term "stainless steel" by itself should not be used as an exact specification of a particular material, for there are almost as many qualities of stainless steel as there are of ordinary steel. In ordinary steel the mechanical properties depend mainly

upon the percentage of carbon which they contain. Thus a steel which is to be easily forged and machined should not contain more than .35 to .40 per cent of carbon. For medium purposes, such as for use in the manufacture of railway tires, springs, hot working tools, etc., the carbon content ranges from .45 to .70 per cent. For edge tools, taps and other cutting implements, steels containing .75 to 1.0 per cent carbon are chosen. In the same way stainless steels may be classified according to their carbon content. At Brown Bayley's steel works the following grades of stainless steel are actually made:

Stainless Iron.—Carbon, less than .10 per cent. Tensile strength, 30 to 70 tons, according to heat treatment. Easy to forge, drop stamp, machine and work cold.

Mild Stainless Steel.—Carbon, .1 to .2 per cent. Tensile strength, 40 to 80 tons. Easily forged under power hammers. Can be hardened and used for cutting tools for soft materials.

Medium Stainless Steel.—Carbon, .2 to .3 per cent. Tensile strength 45 to 90 tons. Too hard to forge by hand, but can be forged into simple shapes under power hammers. Intense air hardening property. Must be carefully cooled. Suitable for sharp and durable cutting tools and for hard-wearing surfaces.

Stainless Steel.—Carbon, .3 to .4 per cent. Tensile strength 50 to 110 tons. Forging qualities, etc., as for medium stainless steel.

Hard Stainless Steel.—Carbon, over .5 per cent. Difficult to forge. Possesses great strength at high temperatures. Stains more easily than milder varieties.

That the mechanical properties of the above different kinds of stainless steel are varied by varying the carbon and not the chromium content may perhaps surprise some readers. Yet, as is shown in the following typical analyses, the chromium percentage can be substantially constant in steels having very different mechanical properties.

	Stainless iron.	Mild stainless steel.	Stainless steel.
Carbon	.07	.15	.37
Silicon	.08	.09	.19
Manganese	.12	.16	.15
Chromium	11.7	11.8	12.0
Nickel	.57	.77	.55

In the course of our previous article it was made apparent that one of the commonest "faults" found with stainless steel was the alleged difficulty of forging it. From what has been said above, it will be understood that in all probability those who found this difficulty were using an unsuitable quality of the steel. It is therefore desirable that the makers' views on this question of forging should be made known to possible users.

Stainless iron, the material with the lowest carbon content, forges, it is stated, as easily as an ordinary steel containing .45 per cent of carbon. As the carbon content increases, forging becomes more difficult until with the hard varieties the operation is almost as troublesome as in the case of high-speed steel. To illustrate the degree to which the carbon content affects the forging qualities, the statement may be quoted that four to six times as much work may be done on stainless iron at one heat as on ordinary stainless steel. Stainless iron may be easily forged by hand into such difficult objects as spurs, but for large articles or for articles made of the harder varieties of stainless steel a power hammer is indispensable.

The temperature at which the forging is carried out requires attention in all the varieties. Between 900 degrees and 1150 degrees C. the ma-

terials can be forged by rapid blows without danger of splitting. But below 850 degrees to 900 degrees C. the deformation becomes more difficult, and if attempted with heavy blows the materials will be unduly stressed or broken.

Stainless steel may be drop-stamped at temperatures between 1000 degrees and 1200 degrees C., but for drop stamping the makers recommend the use of stainless iron, as with it the wear and tear on the dies is much reduced. It is advisable to do as much of the work as possible in preliminary shaping dies.

After forging or drop stamping, the steel, if allowed to cool in the air, will be hard. The hardness attained will depend upon the exact composition of the steel, the temperature at which it is forged or stamped, and the sectional area of the finished product. It may vary from 250 to 500 Brinell, and being of a more or less uncontrollable nature, should not in general be relied upon to secure a required degree of hardness. Whatever hardness is desired should be obtained by a separate treatment.

If the steel is required in a soft condition suitable for machining, it should be reheated to 700 degrees or 750 degrees C. for about an hour and allowed to cool in the air, or quenched in oil or water. It can be made still softer by heating it to 850 degrees or 870 degrees C. followed by very slow cooling, but in this condition, like all annealed steels, it does not machine as smoothly as with the first treatment. It is worth noting that if the steel is subjected to the first treatment it should be magnetic when withdrawn from the furnace. If it is not, its temperature is too high or the soaking has not been sufficiently prolonged.

To harden stainless steel it should be heated to about 920 degrees C., and (1) cooled in the air if the section is small or the degree of hardness required is not great; (2) quenched in oil if the article is of large section and simple in design; or (3) quenched in water if the article is of symmetrical section and able to withstand great stresses. A hardened stainless steel object may be tempered by reheating it to various temperatures. It should be particularly noted that the usual scale of colors is not applicable to stainless steel. The tempering operation is best carried out by means of a salt bath the temperature of which is controlled by a pyrometer. It is a peculiarity of stainless steel that after being hardened it may be reheated to temperatures up to 500 degrees C. without any great effect on its hardness. Within this great range, therefore, it may be tempered to increase its toughness without softening it.

The effect of cold working on stainless steel is another matter requiring the close attention of users. Whatever opinions to the contrary may have been expressed, it is undoubtedly true that the material may be drawn into wire, rolled into sheets, or made into weldless drawn tubes. It is necessary, however, to modify the practice followed in these operations with ordinary steel. For example, the wire rod must not be "patented" before the wire is drawn from it, for the temperature at which patenting is done would make the rod hard and brittle. The air-hardening property of stainless steel must always be borne in mind; to soften the material, whether in the form of wire rod, hot-rolled sheet or hot-rolled tube, it should be reheated to 700 degrees or 750 degrees C., and slowly cooled or quenched. The softened steel, after pickling in the usual acid, is ready for the first cold drawing or rolling process. The intermediate softening is carried out each time it is required by re-heating to 700 degrees or 750 degrees C.

It is to be remembered that all materials distorted by cold working show an increased tendency to corrode. Stainless steel is no exception to this rule, and, consequently, after it has been drastically cold worked, it

should not be put into service until the internal stresses have been relieved by hardening, tempering or other heat treatment. It follows from what has just been said that stainless steel turnings or borings will rust, as will the surface from which the shavings have been removed, if the cut is rough. A surface finished with a fine cut is less distorted, and will, therefore be less liable to rust than one finished with a rough cut. Grinding and polishing by removing the distortion make the surface immune from rusting. It is thus not the case, as has been frequently stated, that the freedom of stainless steel from corrosion is conferred upon it by polishing. A rough surface, if it is not distorted, will be rustless. The grinding of stainless steel also calls for special attention. It is more difficult to grind than ordinary steel of equal Brinell hardness, and if the softer varieties are ground rashly the surface may be distorted and the above-mentioned liability to corrosion set up. On the harder varieties rash grinding may produce soft spots which may stain or crack.

The behavior of stainless steel at high temperatures is characterized by its non-scaling property and by its retention of a considerable proportion of its original hardness and strength. In an ordinary steel after hardening and polishing, reheating produces the well-known temper colors, which are really very thin films of oxidized metal. As the temperature rises a scale of measureable thickness is formed, and at a low red heat the thickness of this scale increases with the time during which the steel is exposed to the corresponding temperature. Experiments have been undertaken to determine the loss of weight by scaling in the case of a variety of steels, including stainless steel. The steels were subjected for sixty-seven hours to a temperature of 700-725 degrees C., then for forty-six hours to 750-800 degrees C., and then for forty-six hours to 800-825 degrees. At the end of this treatment the scale removed from a 5 per cent nickel steel amounted to nearly 22 per cent of the original weight. The scale removed from the stainless steel represented not more than half of 1 per cent of the original weight. Other steels gave intermediate values, the next best result to that of stainless steel being shown by a 25 per cent nickel steel with 2½ per cent loss. A further treatment of twenty-three hours at 835-850 degrees C. caused the stainless steel to lose about 6 per cent by scaling, the 25 per cent nickel steel about 7 per cent and the 8 per cent nickel steel over 30 per cent. The retention of its hardness by stainless steel at high temperatures is shown by the difficulty of forging the higher carbon varieties. As for the retention of its strength, experiments show that high carbon stainless steel has a tensile strength of over 15 tons per square inch at 700 degrees C., as compared with less than 7 tons for ordinary mild steel, and about 9 tons for nickel or nickel-chrome steel. Only one other steel—a special high-speed material—excelled the stainless steel result with 17.7 tons. It is perhaps worth adding that Messrs. Brown, Bayley make certain high chromium, high nickel alloys which cannot strictly be called stainless steels, but which are much stronger at high temperatures either than stainless steel or high-speed steel.

Some remarks on the stainlessness of stainless steel may be added. The degree of resistance of the material to staining and rusting influences varies both with the exact composition of the steel and with its previous heat treatment. It is most resistant when it is in the hardened condition. When tempered at temperatures up to 500 degrees C. the material, as used for the manufacture of table blades, will not stain when exposed under standard test conditions to the action of vinegar. After tempering at 600 degrees C. it is generally found to be stainless under the test conditions. At 630 degrees C. it may or may not be stainless. Tempered at 700 degrees C. or higher, it stains in the vinegar test, but does not rust

under ordinary conditions. If completely annealed at 850-870 degrees C. and cooled very slowly, it stains and rusts, but to a very much less extent than ordinary steel.

Nitric acid, cold or boiling, does not appreciably attack stainless steel, and cannot therefore be employed to etch it. It may, however, be etched by a mixture of nitric and hydrochloric acids, or by hydrochloric acid in which copper sulphate is dissolved. The material does not corrode when it is in contact with other steel. It is, however, attacked in an otherwise non-corroding liquid such as sea water when it is in direct metallic contact with copper or copper alloys, such as gun-metal.

In some quarters, we believe, it is thought that stainless steel has no advantage over a steel containing about 25 per cent of nickel. On the contrary, it possesses a much greater resistance to corrosion. Moreover, it can readily be softened for machining purposes, and subsequently hardened to resist wear or heavy stresses, whereas high nickel steel is very difficult to machine, however, it may be softened, while by no known process applicable to machined articles can it subsequently be hardened.—"The Engineer," November 11, 1921.

A CLOSED-CYCLE AMMONIA MOTOR.

Closed-Cycle Ammonia Motor, Prof. E. Garuffa. Many attempts have been made to produce a "steam" engine employing a liquid vaporizing at a lower temperature than water, and there are numerous patents on such engines, none of which, however, have reached a stage of commercial development.

Of such engines the ammonia motor originally developed by an Italian engineer, B. Caruso, possesses certain features of interest. In this engine an attempt is made to utilize to a very high degree the latent heat of the exhaust vapor, which has never been successfully accomplished in steam engines. The thermal properties of ammonia are claimed to be such as to permit this utilization of latent heat to a greater extent than is possible with water.

The closed cycle operates essentially as follows: In the first place, there is a generator of saturated ammonia vapor doing work in a proper motor (*e. g.*, a turbine) and constantly fed from another generator connected to it. In this second generator the exhaust vapor from the turbine gives up its heat to the liquid, and being condensed to a liquid state goes into an automatic feeder which delivers it to this second generator. Theoretically, under this cycle the only heat that has to be supplied to the ammonia in order to maintain the cycle is the heat corresponding to the work done and the heat necessary to compensate for losses—in particular, those which are not recuperated either before or after condensation. While at the same pressure, which is the maximum pressure of the cycle, the two generators which constitute the funadmental part of the system are, however, at different temperatures, the difference in temperature being the temperature head across the turbine.

Fig. 13 illustrates the manner of actually effecting this cycle. A is a receiver or really a vapor generator heated by some fuel; M is the turbine or other kind of motor which receives the vapor from the ammonia generator after it has passed through the superheater SA, the purpose of which is to utilize the heat in the exhaust from the furnace in the generator A. At B is a tank of liquid ammonia acting as a surface con-

denser; into it is immersed the coil S receiving the exhaust vapor from the turbine and condensing it. The condensate collects in the receiver C, whence it is conveyed by the pump P' to E, which, in its turn, constitutes an automatic feature of the condenser B, the latter acting as the second vapor generator.

The respective positions of the two units, A and B, (which, as has been stated above, act as two vapor generators at different temperatures) are such that they act as two communicating vessels (by pipe P) in such a manner that with equal pressures the levels of the liquid are the same.

Further, between the tank B and the dome D is inserted a special diaphragm *d*, and between the tank B and generator A another diaphragm R. These diaphragms permit the establishment of a uniform pressure throughout D, B and A, this being the maximum pressure of the cycle, at the same time maintaining between A and D and D and B the temperature head necessary for the motor to function.

The vapor which collects in D from surface evaporation of the liquid

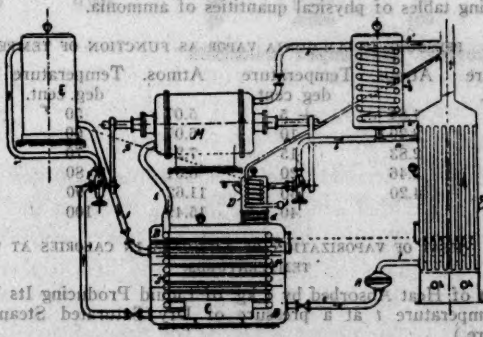


FIG. 13 CARUSO CLOSED-CYCLE AMMONIA MOTOR

in B is in a state of saturation due to its being in contact with the liquid and at a fixed temperature which may vary between 60 and 100 degrees C. in accordance with the existing pressure. It is conveyed by the pump E as saturated vapor, into the generator A and from there, together with the vapor generated therein is sent on its way first to the superheater S_A and eventually to the turbine M.

The purpose of the dome D is to maintain the liquid in B at a constant temperature, preventing the liquefaction of saturated vapor contained therein. To do this the dome is provided with a heating device which may or may not be connected with the furnace in A, and the heat coming therefrom helps to maintain the saturated vapor at a temperature of saturation corresponding to the maximum pressure of the cycle, 94 atmospheres for 60 degrees and 60 atmospheres for 100 degrees C.

It is claimed that the efficiency of such a system is quite high. The heat content of a kilogram of superheated vapor Q is the sum of the heat

content q of the saturated vapor and the superheat in the vapor, the latter being

$$q_1 = c_p\,(t_{sup} - t_{sat})$$

and hence the theoretical quantity G of steam per horsepower-hour in kilograms is

$$G = \frac{632}{Q} = \frac{632}{q + c_p\,(t_{sup} - t_{sat})}$$

(t_{sup} = temperature superheated vapor; t_{sat} = temperature saturated vapor). The actual quantity will be considerably greater, because of the losses in the turbine, generator and condenser, and imperfect efficiencies of these units. Assuming that the heat losses are 10 per cent, efficiency of generator is 75 per cent, efficiency of turbine 70 per cent and efficiency of condenser 85 per cent, we get a total efficiency of 40 per cent and a consumption equal to 2.5 Q.

No statement is made as to whether any such engine has ever been built, and, if it has, what were the actual results of tests on it.

In his work on the ammonia motor Caruso worked out experimentally the following tables of physical quantities of ammonia.

TABLE 3. PRESSURE IN AMMONIA VAPOR AS FUNCTION OF TEMPERATURE.

Temperature deg. cent.	Atmos.	Temperature deg. cent.	Atmos.	Temperature deg. cent.	Atmos.
— 20	1.84	+ 5	5.07	50	19.94
— 15	2.29	10	6.07	60	24.32
— 10	2.83	15	7.21	70	32.45
— 5	3.46	20	8.51	80	40.60
0	4.20	30	11.62	90	50.15
		40	15.49	100	61.30

TABLE 4. HEAT OF VAPORIZATION OF AMMONIA IN CALORIES AT VARIOUS TEMPERATURES.

(Amount of Heat Absorbed by 1 kg. of Liquid Producing Its Vaporization a Temperature t at a pressure of Dry Saturated Steam at that Temperature.)

Temp. t deg. cent.	Calories	Temp. t deg. cent.	Calories	Temp. t deg. cent.	Calories
— 20	327	+ 5	311	50	275
— 15	324	10	308	60	273
— 10	322	15	305	70	268
— 5	319	20	301	80	264
0	315	30	292	90	260
		40	283	100	255

TABLE 5. HEAT NECESSARY TO RAISE THE LIQUID (ANHYDROUS AMMONIA) FROM ZERO TO t DEG.

Temp. t deg. cent.	Calories	Temp. t deg. cent.	Calories	Temp. t deg. cent.	Calories
— 20	17.34	+ 5	4.45	50	51
— 15	18.13	10	9.17	60	63
— 10	8.83	15	14.87	70	75
— 5	4.47	20	19.66	80	88
0		30	29.49	90	105
		40	39.64	100	120

TABLE 6. TOTAL HEAT OF VAPORIZATION OF AMMONIA.
(Sum of the Heats of Tables 4 and 5.)

Temp. t deg. cent.	Calories	Temp. t deg. cent.	Calories	Temp. t deg. cent.	Calories
— 20	344.34	+ 5	315.45	50	326
— 15	337.13	10	317.17	60	336
— 10	330.83	15	319.87	70	343
— 5	323.47	20	320.66	80	352
0	.....	30	321.49	90	365
		40	322.64	100	375

TABLE 7. SPECIFIC VOLUMES OF SATURATED AND DRY AMMONIA VAPOR.

Temp. t deg. cent.	Calories	Temp. t deg. cent.	Calories	Temp. t deg. cent.	Calories
— 20	0.646	+ 5	0.250	50	0.073
— 15	0.525	10	0.221	60	0.051
— 10	0.432	15	0.183	70	0.037
— 5	0.358	20	0.161	80	0.025
0	0.298	30	0.122	90	0.018
		40	0.090	100	0.013

"Mechanical Engineering," November, 1921.

BOOK REVIEWS.

OIL FLOW IN PIPE LINES. BY R. S. DANFORTH, ASSOC. M. AM. SOC. C. E.

This pamphlet contains pressure loss charts for use in solving oil pumping problems in long lines, a viscosity conversion chart and horsepower chart for pumping.

The pressure loss charts may be used in solving a problem for any one of the following quantities, the others being known:

(*a*) Volumetric flow per time unit.
(*b*) Friction pressure loss per unit length pipe.
(*c*) Saybolt viscosity.
(*d*) Baumé gravity.

The charts are derived from formulae based on law of flow of liquids in pipe lines deduced by Messrs. Stanton and Pannell of the National Physical Laboratory of Great Britain. Separate charts are given for standard iron and steel pipe of diameters 2 inches, 3 inches, 4 inches, 6 inches, 8 inches, 10 inches and 12 inches. The charts apply to the flow of any liquid or gas when the inside surface of pipe is of the same roughness as that used in oil work.

The viscosity conversion chart shows the conversion of the times of efflux of the Saybolt Universal, Saybolt Furol, Engler, Redwood and Redwood Admiralty viscometers and Engler specific and absolute viscosity into kinematic viscosity.

The horsepower chart may be used in solving for any one of the following quantities, the others being known:

(*a*) Shaft horsepower.
(*b*) Volumetric flow per time unit.
(*c*) Total pumping head.
(*d*) Mechanical efficiency of pump and drive

JANE'S FIGHTING SHIPS, 1921. (SAMPSON LOW, MARSTON & CO., LTD., LONDON.)

This book is so well known to Naval circles that it requires but little comment. However, this issue is unusually complete and contains not only the lists of ships maintained by each of the naval powers, but also gives the organizations of the various Navies. There is also a list of ships which have been deleted from the previous edition.

WHARF MANAGEMENT, STEVEDORING AND STORAGE. BY ROY S. MAC ELWEE AND T. R. TAYLOR. PUBLISHED BY D. APPLETON & CO., NEW YORK, 1921. CLOTH; 6x9 IN.; 350 PP.; 94 ILLUSTRATIONS; NUMEROUS TABLES AND FORMS. PRICE.

A very complete treatment of a subject of primary importance—if ocean steamship traffic in this country is going to pay. The book presents the many problems to be encountered in the receipt and dispatching of ocean freight, the present methods of handling and stowage at various ports, and suggested improvements of apparent defects.

The volume is especially timely due to the live problem of the disposition of shipping built during the war which is now idle. It should not only prove valuable to those already in the shipping business who desire a broader outlook but should be of special interest to young men in training for this business.

One, in reading the book, is impressed with the necessity for saving time in loading and unloading ships, high cost of labor charges, influence of union labor on installation of mechanical means of handling cargoes, and the large amount of manual labor employed.

Probably the most valuable part of the book to one desiring to make an exhaustive study of this subject will be found in the large number of references and where they can be obtained.

NOTES.

THE TWENTY-FIFTH ANNUAL MEETING of the AMERICAN SOCIETY for TESTING MATERIALS will be held on June 26-July 1, 1922, at Atlantic City, N. J. Headquarters will be at the Chalfonte-Haddon Hall Hotel.

ASSOCIATION NOTES.

ELECTION OF OFFICERS.

Ballots cast for officers of the Society for 1922 were counted at a meeting held at Washington, D. C., on December 30, 1921. The following were declared elected:

President:

Rear Admiral C. W. Dyson, U. S. Navy.

Secretary-Treasurer:

Commander S. M. Robinson, U. S. Navy.

Council:

Rear Admiral B. C. Bryan, U. S. Navy.
Captain Robert Stocker (CC), U. S. Navy.
Captain E. C. Kalbfus, U. S. Navy.
Commander S. C. Hooper, U. S. Navy.
Lieutenant M. R. Daniels, U. S. Coast Guard.

ANNUAL DINNER.

The Society decided by a vote of 157 to 36 that a dinner should be held. The Committee appointed arranged for a dinner on Saturday, March 18, 1922, at the New Willard Hotel, Washington, D. C. These annual events have been delightful occasions in the past and no effort will be spared to ensure that the dinner this year shall excel in every respect those which have preceded it.

FINANCIAL STATEMENTS FOR 1921.

PUBLICATION ACCOUNT.

The following statement shows the cost of producing the JOURNAL of the Society for 1921 and the returns therefrom:

	Returns.	*Costs.*
Printing		$8,041.73
Engraving		1,554.43
Drafting		5.00

Manuscript		1,118.00
Commission		230.60
Postage		122.33
Exchange		.36
		$11,072.45
Dues	$5,801.79	
Subscriptions	1,607.35	
Sales	463.33	
Advertisements	2,396.98	
		10,269.45
Loss on Publication		$803.00

GENERAL PROFIT AND LOSS STATEMENT.

The following statement shows the result of the complete operation of the business during the year 1921:

	Gain.	*Loss.*
Loss on Publication (above statement)		$803.00
General Expense		2,664.92
Current Profit and Loss		1,026.88
		$4,494.80
Interest	$664.93	
		664.93
Net Loss		$3,829.87

STATEMENT OF RESOURCES AND LIABILITIES.

The following statement shows the resources and liabilities of the Society on December 31, 1921, and the net result. It should be understood that the liabilities consist entirely of dues and subscriptions paid in advance, and are not accounts payable by the Society as a going organization. They will be eliminated on the first business day of 1922, and will be credited as profits for that year.

Total Assets January 1, 1921 ..... $20,461.21

	Liabilities.	*Resources.*
Cash		$3,118.81
Accounts Receivable, Advertisements		1,348.61
Accounts Receivable, Dues		814.71
Accounts Receivable, Sales		17.13
Accounts Receivable, Subscriptions		222.70
Army and Navy Club Bond		1,000.00
Certificate of Deposit		1,107.95
Furniture		130.50

Government Bond, 1st Liberty		1,000.00
Government Bond, 3d Liberty		1,000.00
Government Bond, 4th Liberty		2,385.20
Government Bond, 5th Liberty		1,000.00
War Savings Stamps		918.00
Washington Ry. & Elec. Bonds		3,900.00
		$17,963.61
Dues paid in advance	$248.21	
Subscriptions Paid in Advance	1,084.06	
		1,332.27
Net Assets December 31, 1921		16,631.34
Net Loss		$3,829.87

With reference to the loss shown for the year, it is proper to state that $1,332.27 of this amount is caused by an adjustment of the books to show as profit only such transactions as were entirely closed on December 31, 1921. Heretofore all dues and subscriptions received in advance have been credited to income for the year in which they were actually received. This is a book-keeping and not a real loss. Four hundred and twenty dollars was charged for depreciation of securities. This was done to bring the book value of securities to market value on December 31, 1921. They had been carried at cost, with the exception of one previous depreciation charge, for many years. It was considered wise to charge off $709.63 for bad accounts. Six hundred and seventy-three dollars and seventy-five cents of this consists of unpaid dues of members.

This statement has been audited and found correct.

MEMBERSHIP.

It is desired to increase our membership during the coming year, in order that the Society may be of still more benefit to the Service than heretofore. It is requested that members bring to the attention of officers, and especially the younger officers, who are not members, the advantages of the Society and extend to them a most cordial invitation to join with us. It is, perhaps, too well known to require restatement that the Society is conducted solely for the professional benefit of the Service, and that no financial profit is sought.

The following members and associates have joined the Society since the publication of the last JOURNAL.

MEMBERS.

Brantly, Neill D., Ensign, U. S. Navy,
Deming, Raymond A., Lieutenant, U. S. Navy.
Devine, A. D., Lieutenant, U. S. Navy.
Grove, M. H., Ensign, U. S. Navy.
Osborn, Donald R., Jr., Ensign, U. S. Navy.
Searles, Paul J., Lieutenant (CEC) U. S. Navy.
Tompkins, J. T., Captain, U. S. Navy.
Wood, Gerald H., Lieutenant, U. S. Navy.
Wurtele, A. R., Lieutenant, U. S. Navy.

ASSOCIATES.

Callahan, Joseph G., S. S. Steelmaker, Isthmian S. S. Lines, 39 Cortlandt Street, New York City.

Dubertret, Paul, Chief Engineer, Cie Electro Mecanique, 12 Rue Portalis, Paris, France.

Harsant, W. J., Chief Engineer, Pittsburgh Steamship Co., Rockefeller Building, Pittsburgh, Pa.

Matthews, John B., Consulting Engineer, 24 California St., San Francisco, Calif.

Please do no fail to keep the Secretary advised as to your ADDRESS in order to ensure delivery of your JOURNAL.